SAE JOURNAL

August, 1949

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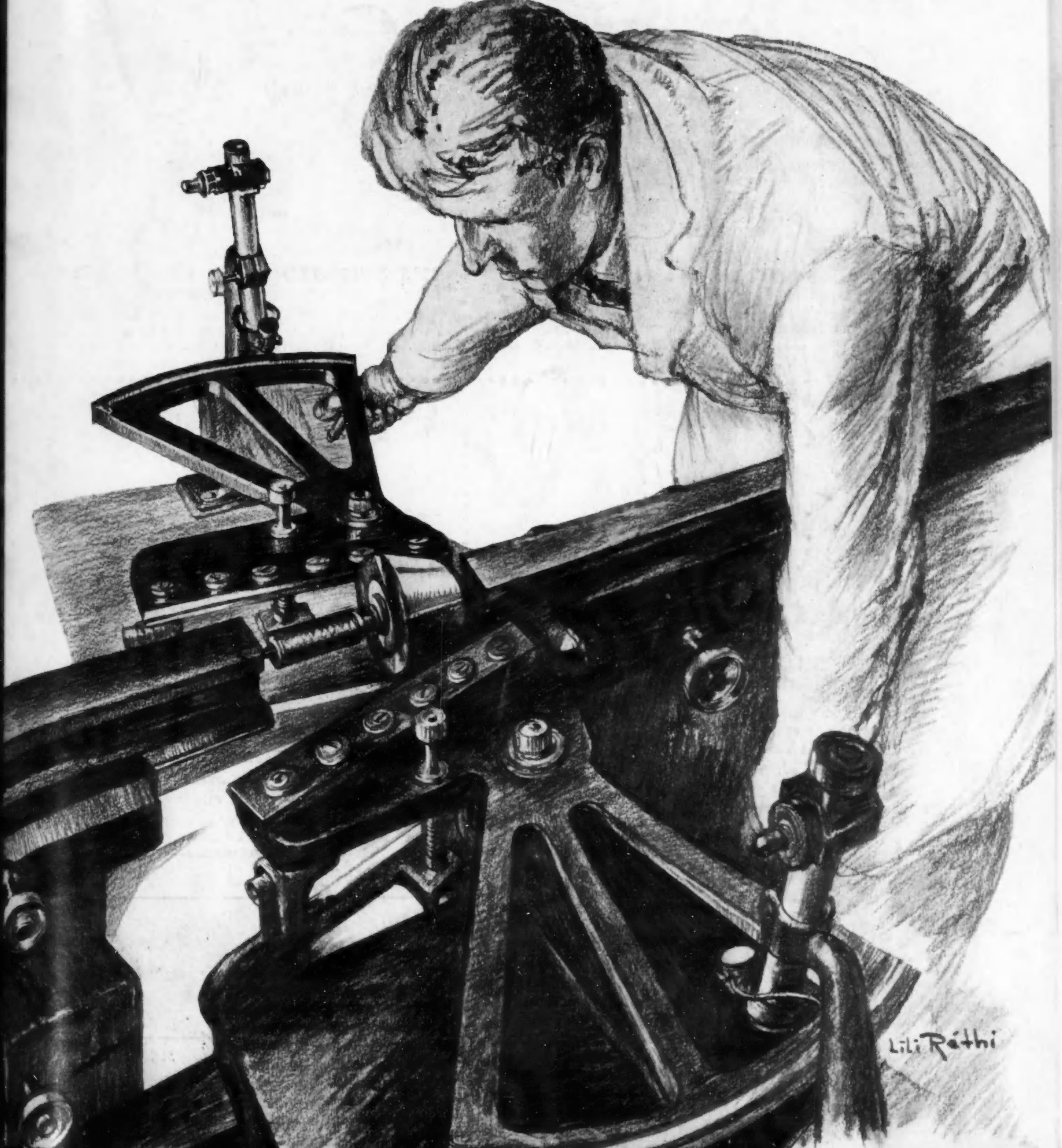


THE PERFECT CIRCLE

Rumor Page

IT'S RUMORED THAT: ***Front-seat television soon may be prohibited by law!***

RIGHT! The National Committee for Traffic Safety, representing 85 organizations, has asked lawmakers in all states to ban front-seat television in autos.

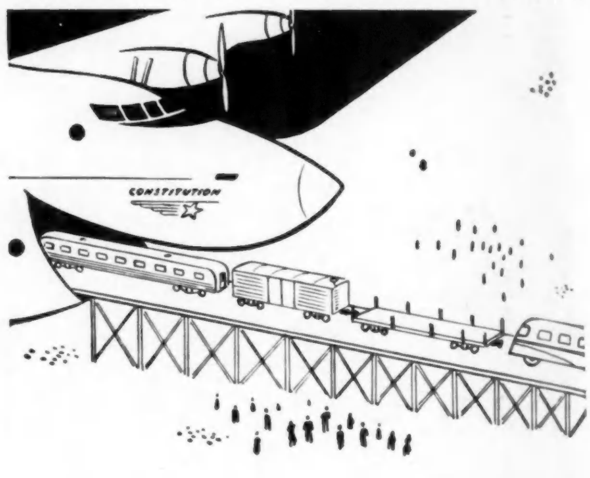
*Contributed by Betty L. MacQueen, Spencer, W. Va.**

IT'S RUMORED THAT: ***There really is a left-handed monkey wrench in existence!***

YEP. . .'S A FACT! . . . . A wrench recently discovered (made on an 1871 patent) has a *right-hand thread,* making it easier for a left-handed man to use it!

IT'S RUMORED THAT: ***There's a piston ring that "reduces" abrasives in engines!***

THAT'S RIGHT! The New PC Ring, plated with solid chrome, has a surface *so hard* that it wears down abrasive materials into smaller, smoother particles. Thus, the PC Chrome Ring actually reduces wear on other rings and cylinders! But write us and get all the facts on the ring with a wear rate 80% below unplated rings!



IT'S RUMORED THAT: ***The U. S. Navy's "Constitution" could hold a Pullman Car, boxcar, flatcar and passenger bus—all at once!***

IT CAN! To give you another idea of the "Constitution's" size . . . it weighs 92 tons, has a wing span of 189 feet, is 156 feet long, and can carry 180 passengers (including a crew of 12).

***Perfect Circle pays $50.00 for any Rumor accepted for this page. None can be returned or acknowledged, and all become PC's property. Send yours to Perfect Circle Corporation, Hagerstown 9, Indiana.**

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AUGUST

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BUILDERS OF THE BASICS OF
BETTER MOTOR VEHICLES
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Bendix* Brakes for Buses, Trucks, and Passenger Cars
Bendix-Weiss Constant Velocity Universal Joint
Centermount Emergency and Parking Brake for Buses and Trucks
Bendix Vacuum-Power Gear Shifter
DEALERS AND TRUCK OPERATORS AGREE...

Selection of Steel for Automobile Parts

What Engineers Should Know Today About Hardenability-Band Steels

Part I—Introduction to Hardenability

This is the first of a six-part report issued by the SAE Iron and Steel Technical Committee that will appear serially in succeeding issues of the SAE Journal. This report was prepared at the request of the SAE Iron & Steel Technical Committee's Division XVIII, Hardenability Publications. Part I was prepared by Joseph Geschelin, Chilton Co., from material provided by the Committee's Division III, Hardenability Bands. Parts II-VI were prepared by A. L. Boegehold, Research Laboratories Division, GMC.

BASED on many years of experience the selection of steels for the major parts of automotive vehicles has been developed by a method of trial and error with past experience—good or bad—as the final criterion of the material thus specified.

Because the factors involved in a steel specification for a given part are so complex and because the element of cost is so important, the metallurgists of the industry have been seeking some simple solution that would aid the engineer in making a proper choice and at the same time make it possible for the engineer and metallurgist to get together on a common footing. Obviously unless the designer is familiar with latest developments and is sold on their application, the practical solution of the problem will be unduly delayed.

One of the earliest disclosures of the vital role played by the Jominy end-quenched hardenability specification was presented about 11 years ago; but because of its novelty it excited the attention only of metallurgists. More recently A. L. Boegehold, of the General Motors Research Laboratories, presented a complete discussion of the state of the art in a paper entitled "Selection of Automotive Steel on the Basis of Hardenability" (see SAE Transactions—Vol. 52, No. 10, October, 1944).

On the basis of extensive experience gained during the war, leading metallurgists of the industry are agreed that the time-honored method of specifying structural steels on the basis of chemistry, modified by stated physical properties, not only is wasteful so far as materials cost is concerned, but ofttimes ineffective in producing the desired results. Surprising as it may sound to an engineer, it is now found that under certain conditions the engineer can get the same results from steels of lowest cost as from the most expensive alloys, so long as the proper heat-treatment is applied in each case.

What can be accomplished through the use of the so-called "H-band" series of steels now preferred by the metallurgists was summarized in a paper "Present Day Approach to the Choice and Application of Automotive Steels," presented by W. E. Jominy, staff engineer, Chrysler Corp., at the 1948 SAE Annual Meeting. Among the factors that Jominy said should be involved in the selection of automotive steels are:

1. There is not much difference in the engineering properties of alloy steels so long as they are properly heat-treated.
2. Proper heat-treatment involves obtaining tempered martensite at the locations of high stress.
3. The greatest difference between alloy steels is the difference in their hardenability.
4. Since to minimize distortion it is necessary to quench most important parts in oil, the amount of hardenability to harden the part in oil determines the minimum acceptable hardenability that the steel may have. If distortion is not a problem, enough hardenability to harden in water is all that is required.

It has been demonstrated conclusively that hardenability—based upon the range of hardness values determined by the engineer and metallurgist working cooperatively—is a major criterion in the selection of the right steel at lowest cost for any given application. Steels manufactured to hardenability bands are within closer limits, as measured by performance in fabrication and heat treatment, than steels made to chemical limits alone.

By starting the use of "H-band" steels without further delay, the engineer will have a simple but effective tool at his command, permitting him to specify the minimum and maximum limits of hardenability for a given steel, and thereby providing better control of hardness variations from lot to lot of purchased steel.

(In the succeeding articles of this series a review of some of the fundamentals of the hardenability of steel will be found together with some of the latest data on hardenability, arranged to show the advantages to be gained by the use of hardenability specifications. Hardenability may not tell everything there is to know about a steel, but it does tell more than any other test known today.)

Copies of the complete six-part series on Hardenability (SP-59) are available from Special Publications Department, Society of Automotive Engineers, 29 West 39th Street, New York 18, N. Y. Price: $1.25 per copy to SAE members, $2.50 to non-members. Quantity prices on request.

TORQUE
CAR

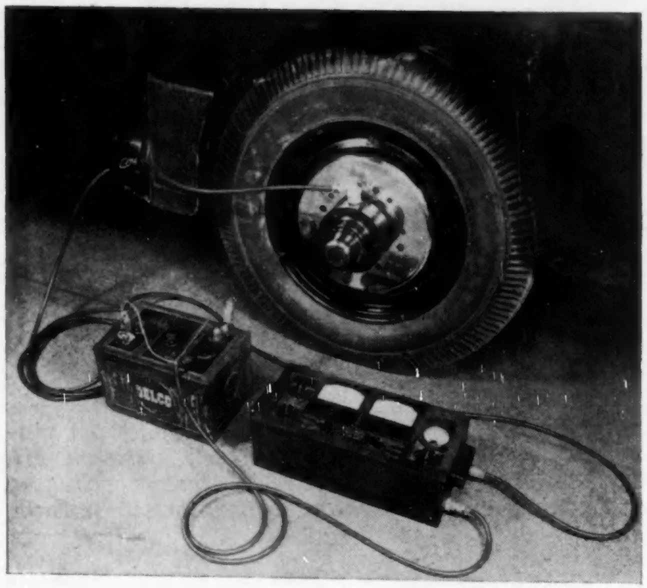

Fig. 1—Passenger car torque wheel and allied equipment for determining wind and rolling resistance

TORQUE wheels make possible a test procedure for measuring car wind and rolling resistance that overcomes objections to previous methods. This technique reveals that current American passenger cars fall into a fairly narrow wind and rolling resistance range, averaging 31 hp at 60 mph.

The device consists of torque-measuring elements installed in the rear wheels. Wheels containing these elements are transferred from one car to another and necessary tire changes made. Chassis losses—except those from front wheel bearings—do not affect the torquemeters because of their location.

Instrumentation, self-contained in the test car, makes the unit highly flexible. It also eliminates the hazards of high-speed tow tests and the possible influence of air flow of a car in the immediate vicinity. Experience also has proved that instrumentation, giving instantaneous torque readings, reduces test time to a fraction of that required by earlier coasting deceleration and drawbar scale towing techniques.

Such coasting deceleration test conditions as extremely low wind velocity and low traffic density do not limit torquemeter tests. The new procedure also overcomes towing method disadvantage of confining tests to summer months, when ambient air temperatures are high, to keep differential lube viscosity within narrow limits for consistent results.

Equipment Installation

Torquemeters are installed on the car by first bolting an adapter plate to the hub with the wheel bolts. Each of the half dozen or more wheel bolt circles used in the industry requires a different adapter. The torquemeter then is bolted to a special wheel rim.

The torquemeter forms a part of an electrical bridge circuit, which is unbalanced proportional to relative angular movement between the hub adapter and special wheel rim. Electrical connections are carried through a slip ring assembly. Torque can be observed on indicating meters or recorded on an oscillograph. A tachometer built into the unit permits observation or recording of both wheel rpm and torque.

Fig. 1 shows the torquemeter installation, including the box carrying the indicating meters.

Test Procedure

The wind and rolling resistance test is conducted by operating at constant speed on a level straightaway, driving in both directions to minimize wind effect, and observing wheel torque and speed. Torquemeters should be installed on both wheels for best results, although we ran many successful road-load tests using only one.

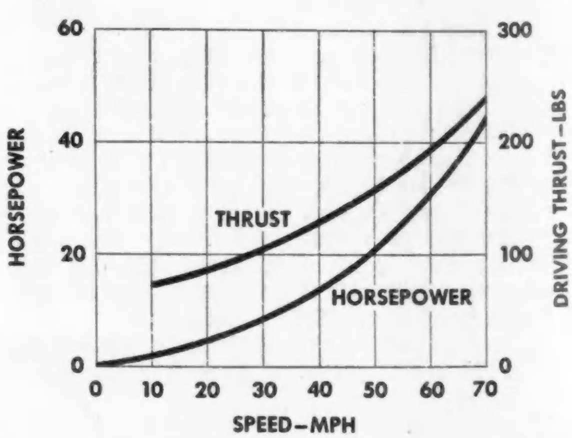

Fig. 2—Average road-load wind and rolling resistance for 20 passenger cars of 1947 and 1948 vintage

WHEELS SPEED ROAD-LOAD TESTS

BASED ON PAPER* BY

K. A. Stonex

Head, Technical Data Department
General Motors Proving Ground

This procedure makes it necessary to measure wheel rpm, so that results can be expressed in terms of wheel horsepower. Variation in rolling radius and standing height, and the uncertainty of their values at any time, make direct conversion from wheel torque to driving thrust generally inaccurate.

The ease of running these tests has modified our conception of the precision required. Time needed for the older techniques made us feel that no road-load test was worthwhile unless conditions were ideal and most precise results were obtained.

* Paper "Passenger Car Wind and Rolling Resistance," was presented at SAE National Passenger Car, Body, and Production Meeting, Detroit, March 10, 1949. (This paper is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

Then it was considered entirely reasonable to wait days, or even weeks, for a perfectly calm early morning or evening for a test. Now we realize it is much more important to have several good tests under reasonably good conditions. And results are perhaps more valuable if they represent operating conditions over the range the customer will meet a fair part of the time.

Fig. 2 summarizes a survey of road-load wind and rolling resistance, measured with torquemeters, of 20 1947 and 1948 passenger cars, ranging from the smallest to the largest. It gives the average horsepower required to overcome wind and rolling resistance. This in turn has been converted to equivalent driving thrust.

As Fig. 3 shows, the cars have nearly the same

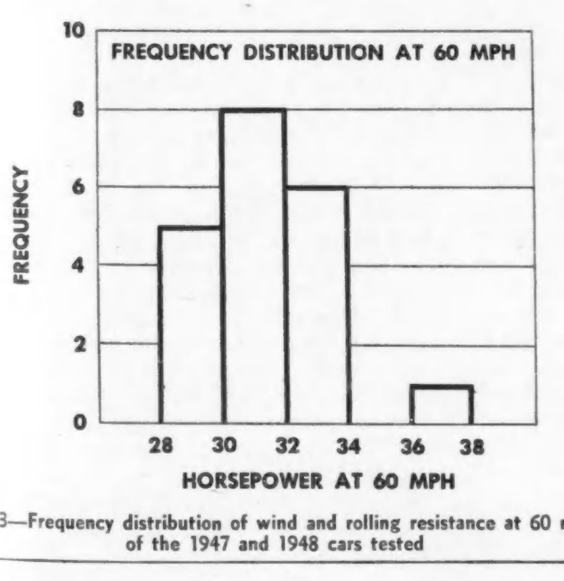

Fig. 3—Frequency distribution of wind and rolling resistance at 60 mph of the 1947 and 1948 cars tested

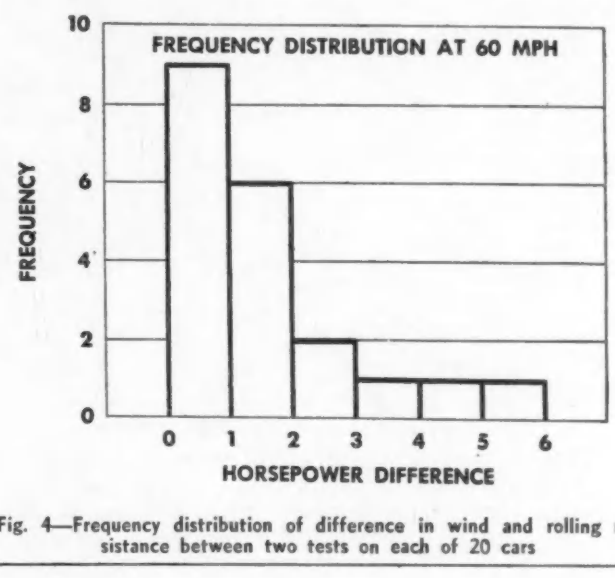

Fig. 4—Frequency distribution of difference in wind and rolling resistance between two tests on each of 20 cars

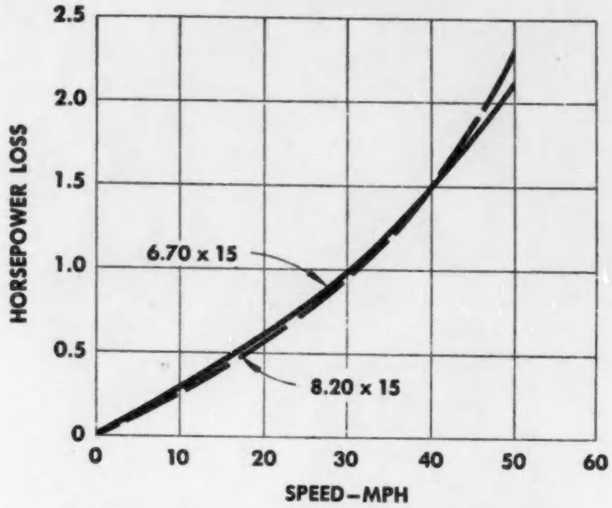

Fig. 5—Road-load tire losses of 6.70 x 15 and 8.20 x 15 sizes

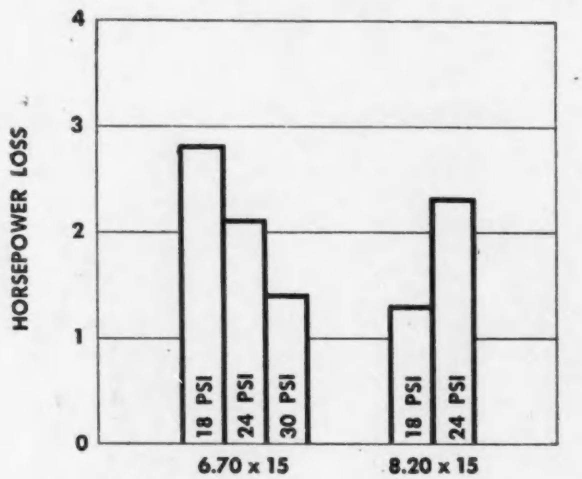

Fig. 6—Comparing road-load tire losses of 6.70 x 15 and 8.20 x 15 tires at varying inflation pressures

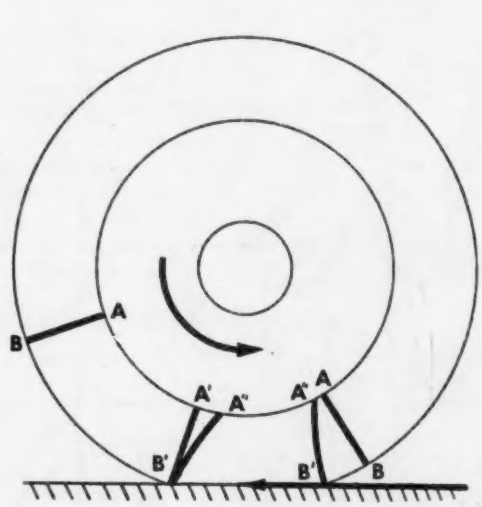

Fig. 7—This diagram partially explains tire loss. Tire distortion permits some tire windup so that the wheel rim gains some distance. While the point B′ moves through the contact surface, wheel angular travel exceeds this by A′A″

wind and rolling resistance. There is hardly more difference between the smallest and the largest than there might be between several repeat checks on the same car over a period of several months, and under a variety of weather conditions.

Fig. 3 shows the narrow variation range between the cars. This frequency diagram of road-load horsepower at 60 mph reveals that all cars but one fall between 28 and 34 hp, and that the average is 31 hp.

Fig. 4 is a frequency diagram of the difference between pairs of tests on the same car. The average difference is 1.6 hp. In 25% of the cases it is more than 2 hp.

Tire Losses Determined

Another interesting item disclosed in torquemeter tests is power transmission losses in the driving wheel tires. They amount to about 2 hp at road-load in both 6.70 × 15 and 8.20 × 15 extra low pressure tires at the usual 24 psi inflation pressure. Large inflation pressure changes produce a small effect on the road-load power losses in the tire.

Fig. 5 shows the comparative road-load losses of the 6.70 × 15 and 8.20 × 15 tires as a function of speed. There is no difference between tire sizes.

Comparative effect of inflation pressure on the two tire sizes at 50 mph road-load is shown in Fig. 6. For the 6.70 × 15 tire, the losses are about 2.8 hp at 18 psi, 2 hp at 24 psi, and 1.5 hp at 30 psi. For the 8.20 × 15 tire, the losses are 1.2 hp at 18 psi and 2.2 hp at 24 psi.

Reversal of effect of increasing inflation pressure in the two tire sizes is not clearly understood. It may be caused by relative changes in standing height and circumferential stiffness. The rim standing height—height of the deflected tire section—changes by nearly 13% from 18 to 30 psi in the 6.70 × 15 tire. But the deflected tire standing height in the 8.20 × 15 size changes only about 5%.

No observations were made of circumferential stiffness. Only one set of tires in each size was observed. Variation among tires of the same make or between different makes was not checked.

Tire Loss Explained

The nature of tire loss is not well understood. Fig. 7 suggests the mechanism of a part of it. This is a diagram of a wheel and tire under driving torque.

Briefly the hypothesis is that forces are applied to the tire only at the road surface and circumferential tire deflection is localized in this area. Since car weight pins the tire surface to the road, tire deflection or distortion permits an increment of tire windup, and the wheel rim must continuously gain some increment of distance if deflection exists.

Car speed will depend on the rate at which some point B′ on the tire periphery moves through the contact area; and the wheel angular travel during the same period will be greater by an amount A′A″, which is the deflection. Torque and consequently road thrust are independent of the distortion. But wheel power is proportional to wheel speed. And under distortion, power input of the tire will proportionately exceed the output.

This is a power loss in the tire. The amount will depend on torque, height of the tire section, and circumferential tire stiffness.

Quality Control—for More Production, Less Cost

BASED ON PAPER* BY

J. N. Berrettoni

Dr. J. N. Berrettoni & Associates

INSTALLATION of a quality control system in an automotive plant results in many benefits:

1. More efficient inspection at lower cost, because the inspector is adequately equipped with appropriate gages; he is trained to understand and use the sensitive graphical tools of quality control; and he acquires confidence in making correct decisions quickly; for quality control, in requiring periodic samples, provides him with a historical commentary on the part being produced, from day to day, month to month, and year to year.

2. Live and dynamic records of quality are provided, which are a constant source of valuable information. For instance, these records help the engineering department to formulate a compromise between functional and manufacturing tolerances and the foundry to determine the desired level of physical and chemical properties of metal and sand.

3. Significantly lowered operating costs, because of the elimination of selective fitting, increased control over and longer life for tools and dies, reduced machine and foundry scrap, and many, many others.

4. Increased assembly output, because fewer defective parts enter the assembly line, which is thus assured a continuous quantity of parts of generally better quality.

5. Increased effectiveness of inspection and supervision, because of the careful allocation of responsibilities from the quality control supervisor on through the various inspectors.

Integral parts of statistical quality control systems are: the frequency histogram and curve, control charts, and sampling plans. Each of these tools of quality control should be thoroughly understood by those supervising its actual operation.

Frequency Histogram and Curve

The frequency histogram and curve are merely graphical pictures that indicate at a glance whether engineering specifications are being held. The following examples show how they are developed and what information can be derived from them:

1. Consider the case of a rocker arm pin with a specified outside diameter of 0.8105 ± 0.0005 in. First, 56 pins, taken at random from a tray of 5000, were measured and grouped according to their respective sizes. Then a series of rectangles was drawn (as shown in Fig. 1), the height of each rectangle being proportional to the number of pins concentrated at that measurement. This series of rectangles is known as a frequency histogram. A smooth curve is then drawn through the tops of the rectangles (also shown in Fig. 1). This is called the frequency curve. It is the histogram that would be obtained if not only all 5000 pins were measured and grouped according to size, but also millions and millions of pins. A frequency curve based on 50-100 in a random sample is, however, a good indication of the variability of the lot, regardless of the lot size.

The frequency histogram and curve throw light on three important aspects of engineering blueprints, namely:

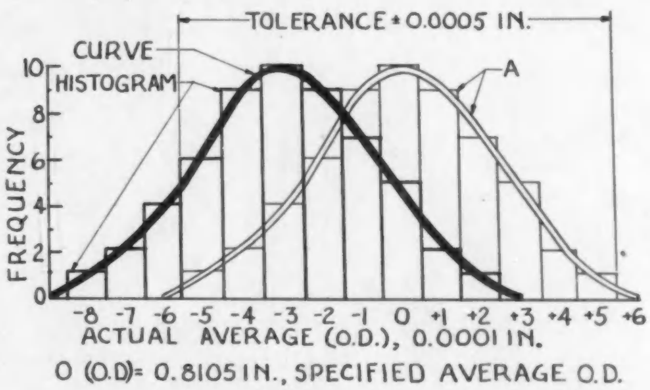

Fig. 1—Shifting frequency histogram and curve by positive 0.0003-in. adjustment of grinding wheel results in producing rocker arm pins at nominal or specified average, with overall variability coinciding with desired tolerance of ± 0.0005 in.

* Paper "Installation of a Quality Control System in the Automotive Plant of Minneapolis-Moline Power Implement Co.," was presented at the SAE National Production Meeting, Cleveland, Oct. 21, 1948. (This paper is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

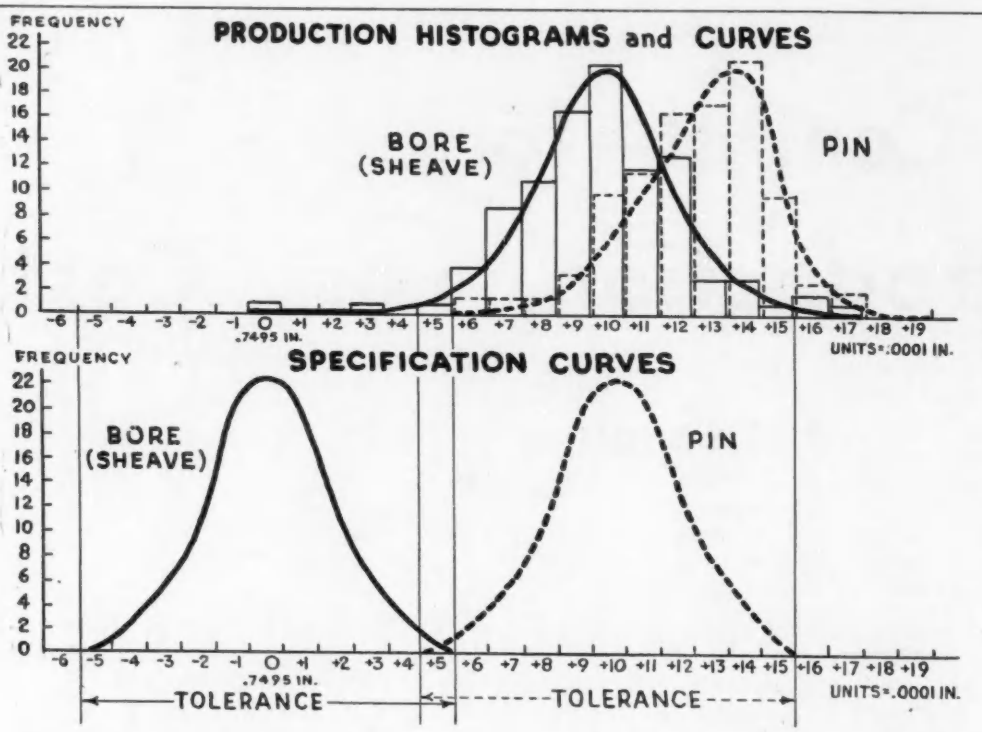

Fig. 2—Comparing actual production histograms and curves with specified engineering curves of bore (sheave) and pin sizes refutes contention that breakage of sheaves is due to too large pin or too small bore, or both

(1) They provide an immediate comparison of the actual average O.D. with the desired nominal or specified average. Fig. 1 shows that there was a negative difference of 0.0003 in.

(2) They show at a glance that the pins varied from −0.0008 to +0.0002 in., whereas the specifications called for ±0.0005. Since 7 of the 56 pins were outside the specified limits, approximately 12.5% of the pins were defective. We note further that the overall variability of −0.0008 to +0.0002 in. corresponds to the overall specified variability of ±0.0005.

(3) The form of the curve indicates that production was symmetrical, that is, there was no tendency to produce to the high side and taper off to the low or to concentrate production at several sizes.

A study of the above factors indicated that the operator could produce pins to specification if he adjusted his grinding wheel to a positive 0.0003 in. His actual average would then equal the nominal or specified average and his overall variability would coincide with the desired tolerance of ±0.0005 in. This adjustment has the effect of shifting the frequency histogram and curve to the right by 0.0003 in., as shown by *A* in Fig. 1.

2. The explanation given for the breakage of sheaves being encountered when a pin was press fitted into a bore (in the sheave) was that the pins were too large or the bore too small, or both. The breakage was about 10%. This contention was investigated by means of the frequency histogram and curve. Fig. 2 shows the results. Bore sizes were definitely larger then the upper blueprint limit and tended to coincide with the O.D. sizes of pins. Thus, the sizes of pins and bores threw doubt on the contention that pins were too large or bores too small. Further investigation showed that the breakage was actually caused by not oiling the pins before pressing. The machine operations were also subjected to quality control techniques to eliminate the bias of the O.D.'s to the high side and the approximately 100% defective work in the bore sizes.

3. Fig. 3 shows the frequency histograms and curves for the O.D. of pistons before and after corrective measures were adopted. The "before" presentation shows two points of concentration and relatively large overall variability. Very few piston diameters are produced at the desired nominal

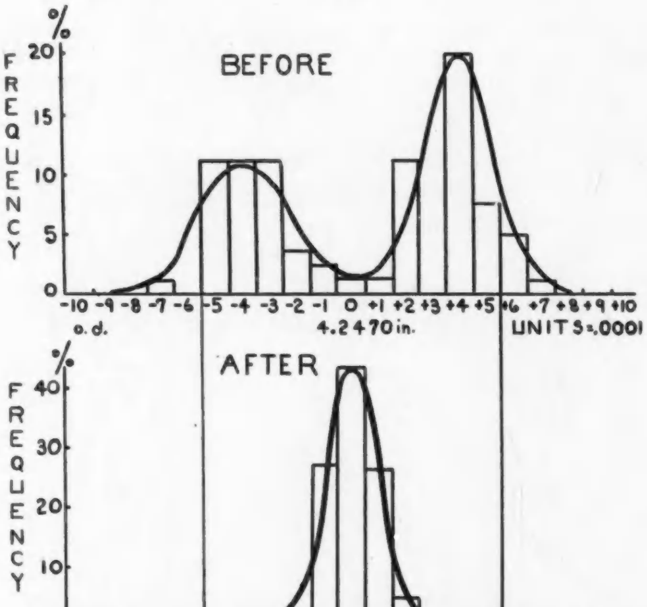

Fig. 3—Frequency histograms and curves of O.D. of pistons before and after corrective measures were instituted

dimension. The "after" presentation shows the effects of instituting corrective measures with respect to coolant, grinding wheels, dressing the wheel, and so on. The tolerance of the pistons is now within ± 0.0003 in.

Control Charts

Only two control charts—the average or $\overline{X}$ chart and the range or R chart—will be discussed. Fig. 4 shows these charts for gear blanks with a specified width of 1.265 ± 0.005 in. All measurements are given as plus or minus, in thousandths of an inch, from the nominal of 1.265 in. This is the same as setting a dial indicator for 1.265 at the zero value and reading the measurement as plus or minus the number of thousandths from zero.

Each dot in Fig. 4A represents the average width of five consecutive gear blanks taken directly from the automatic machine. The average is equal to the sum of the widths of the five consecutive pieces divided by five. Each corresponding dot in Fig. 4B is the difference between the lowest and highest values in the sample of five.

The nominal dimension of the $\overline{X}$ chart is identical with the engineer's nominal dimension of 1.265, but the nominal dimension for the R chart corresponding to the engineer's tolerance of ± 0.005 in. is 0.0039, as calculated for a sample of five by the quality control technician. The zero and the 0.0039-in. values are marked on the charts as dotted lines. The solid parallel lines on each chart are control limits. These are also calculated for a given sample of five and a tolerance of ± 0.005 in. For the X chart these control limits are at the ± 0.0022-in. points, and for the R chart they are at the 0 and + 0.0082-in. points. These control limits are essentially the tolerance of averages and ranges, which are derived mathematically from the engineering specifications.

If both the average values of the $\overline{X}$ chart and the range values of the R chart fall within their respective control limits and both vary about their nominal dimensions (the dashed lines), then the individual widths of each gear blank will meet the specified 1.265 ± 0.005 in. If an average value plots outside the control limits there must be a *real* shift from the desired engineering nominal of 1.265 in. Similarly, if a range value plots outside its upper control limit (the lower control limit is not relevant because it is zero), then there is a *real* shift from the quality control nominal range of 0.0039 in., which reflects the given tolerance of ± 0.005 in. In either case, in all likelihood, individual gear blanks will be or are being produced outside the specified 1.265 ± 0.005 in. This is because the average value indicates whether production is at too high or too low a level and the range value indicate whether successive pieces are being produced with too great a variation. The higher the range value the greater the variability. The result of instituting corrective measures is shown in Fig. 4. In the "before" presentation, samples of five yielding out-of-control values for $\overline{X}$ and R (or both) contain at least one gear blank exceeding the specification, as shown in the tabular presentation. In the "after" presentation, all gear blanks are within 1.265 ± 0.005 in.

A sample of only five pieces is used because the effect of causes creating defective work is relatively more pronounced in a small than in a large sample. The average and range values are plotted opposite

SPECIFICATION

NOMINAL = 1.2650 in.

0 = 1.2650 in. UNITS = .001 in.

SAMPLE SIZE 1	2	3	4	5	$\overline{X}$	R	
-1.5	-16.5	-3.5	+3.0	0	-3.7	19.5	BEFORE
-6.0	+2.5	+5.0	0	-3.0	-0.3	11.0	
0	-1.0	-2.0	-15.0	-0.5	-3.7	15.0	
+3.0	-3.5	+1.0	+11.0	+2.0	-2.7	14.5	
+1.0	+1.0	-11.0	0	-3.0	-2.4	12.0	
+2.0	+3.0	-1.5	0	-3.5	0	6.5	
+2.0	-12.0	+1.0	+2.5	-2.0	-1.7	14.5	
+0.5	-0.5	+3.0	+2.0	+3.0	+1.6	3.5	AFTER
+1.5	+0.5	-1.0	+0.5	+0.5	+0.4	2.5	
-1.0	+2.0	+2.0	+2.0	+5.0	+2.4	6.0	
0	+0.5	0	0	+0.5	+0.2	0.5	
0	0	0	-0.5	0	-0.1	0.5	
0	+1.0	0	+2.0	-2.0	+0.2	4.0	
+1.0	+1.0	-1.0	+1.0	0	+0.4	2.0	
-1.0	-1.0	+1.5	0	+1.0	+0.1	2.5	
+0.5	+1.5	+2.0	0	+2.5	+1.2	2.5	
+1.0	-3.0	+2.5	-0.5	+3.0	+0.6	6.0	

TOLERANCE = ±.005 in.

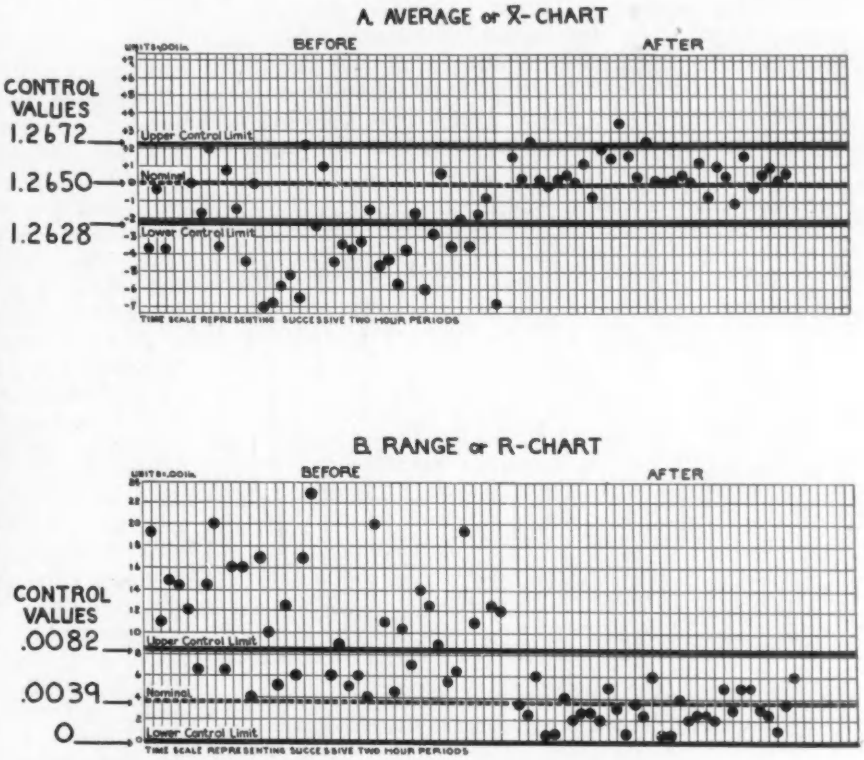

Fig. 4—Average or $\overline{X}$-chart and range or R chart based on engineering specifications for width of gear blanks

Table 1—Series of Sampling Tables—Single Sampling

AOQL = ½%

Lot Size	Sample Size, n	Acceptance No., c
100–950	73	0
951–3000	165	1
3001–7500	270	2
7501–15,500	385	3
15,501–31,000	505	4
31,001–58,000	630	5

AOQL = 1%

Lot Size	Sample Size, n	Acceptance No., c
100–450	36	0
451–1500	84	1
1501–3750	135	2
3751–8000	190	3
8001–15,000	250	4
15,001–28,000	315	5
28,001–50,000	380	6
50,001–85,000	445	7

AOQL = 2%

Lot Size	Sample Size, n	Acceptance No., c
50–250	18	0
251–750	42	1
751–1800	68	2
1801–4000	97	3
4001–7500	125	4
7501–14,000	155	5
14,001–25,000	190	6
25,001–42,000	220	7
42,001–70,000	255	8

AOQL = 3%

Lot Size	Sample Size, n	Acceptance No., c
50–150	12	0
151–500	27	1
501–1500	45	2
1501–2500	64	3
2501–5000	84	4
5001–9500	105	5
9501–17,000	125	6
17,001–28,000	145	7
28,001–45,000	170	8
45,001–75,000	190	9

their time of inception because, as soon as control is under way, any dot that later plots out of control will indicate the approximate time when the causes producing the defective work have intruded. This in turn leads to ease in detecting and eliminating the causes, since the time element assists in segregation of operators, shifts, materials, and the like. Further, if a series of dots plots in such a manner as to indicate that defective work is soon to be obtained (for example, a trend pattern due to tool wear), then corrective action by the foreman may be put into effect before obtaining the defective parts. On the other hand, if dots continue outside of the control limits when everything possible has been done to bring about control, this shows there

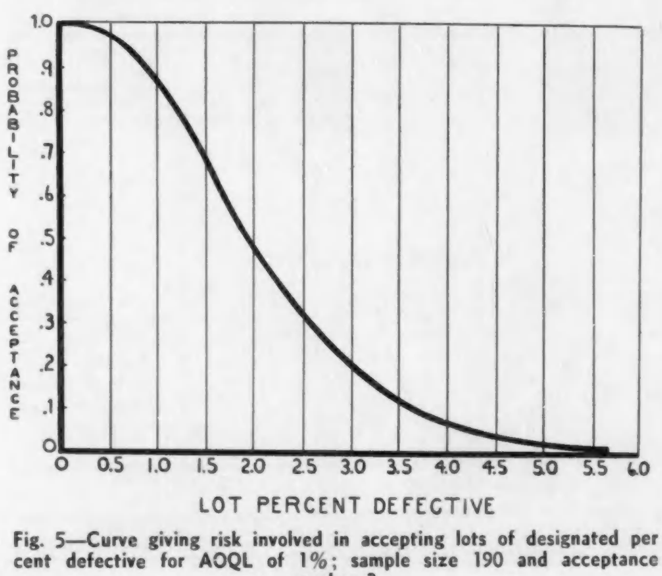

Fig. 5—Curve giving risk involved in accepting lots of designated per cent defective for AOQL of 1%; sample size 190 and acceptance number 3

must be a change made in either the specifications or the production technique.

Sampling Plans

In many instances of industrial manufacturing an arbitrary number of defectives in an arbitrary sample (a spotcheck) is used to judge quality of a lot, that is, to accept or reject the lot. A number of scientific sampling plans have been devised to replace such unscientific methods. One by the author and D. Greb of Minneapolis-Honeywell Regulator Co. will now be described with the aid of Table 1.

1. Selection of average outgoing quality limit (AOQL): Theoretically, the selection of AOQL is a highly complex matter involving a balancing of cost of inspection with all other aspects of cost in the assembly line and quality of the product. Practically, however, a highly experienced engineer can easily stipulate in a matter of moments what value of AOQL applies to a particular part.

Consider, for example, a lot of value guides received by Minneapolis-Moline. The engineering department issues a directive that the maximum average per cent of defective guides to be tolerated over all lots in assembly is 1%. To the receiving inspector this means that the average outgoing quality of guides in all lots leaving the receiving inspection department and entering the assembly line is not to exceed the upper limit of 1%.

2. Determination of lot size: When the AOQL has been designated, the size of the lot is obtained by use of the receiving inspection report.

3. Selecting the sample size and acceptance number yielding the AOQL at minimum cost of inspection: The next step is the determination of the sample size n and the acceptance number c from Table 1. The acceptance number is the highest number of defectives in the sample permitting the acceptance of the lot.

For example, for a lot of 5000 guides, assuming an AOQL of 1%, the table shows that $n = 190$ and $c = 3$. Therefore, 190 valves are inspected from the lot, and if no more than 3 are defective, the lot is accepted and sent to the assembly line or stock. If there are more than 3 defectives, the lot is rejected and must be either sorted by the purchaser or returned to the supplier for sorting. Such a sampling plan thus assures a quality not exceeding an AOQL of 1% over a series of lots, obtained at minimum cost of inspection.

4. Risk factor of sampling: In any attempt at deducing the quality of a lot by means of a sample, there always remains the risk of rejecting good lots and accepting bad ones. The lot of 5000 guides may contain only 4 defectives (0.08%) and still it is possible in a sample of 190 to obtain these defectives and, as a consequence, reject a good lot. On the other hand, the lot may contain 250 defectives (5%), and it is possible in a sample of 190 to obtain only 3 defectives and, as a result, accept a highly undesirable lot. The risk factor for any per cent defect lot can be obtained from Fig. 5. For example, for an AOQL of 1%, sample size of 190, and acceptance number of 3, the probability of accepting a lot of 0.08% defectives is 99.999% (the risk of rejection is 0.001%) and the probability of accepting a lot of 5% defectives is 2%.

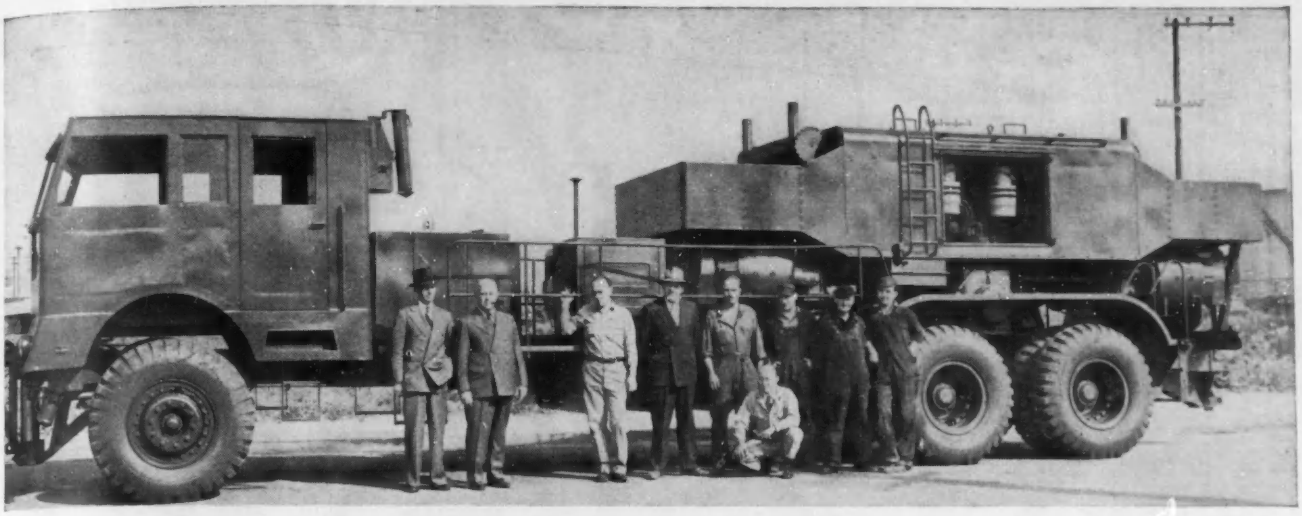

Fig. 1—The 60-ton field dynamometer truck is 44½ ft long, 10 ft-10 in. wide, and 11 ft-4 in. high. Used to determine drawbar effort and traction resistance of military vehicles, it can absorb a 60,000-lb drawbar pull at up to 4 mph

Largest Field Dynamometer

EXCERPTS FROM PAPERS* BY

George U. Brumbaugh

Peterbilt Motors Co.
(Formerly with Knuckey Truck Co., Inc.)

J. E. Wilson

Locomotive & Car Equipment Division,
General Electric Co.

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General Engineering and Consulting Laboratory,
General Electric Co.

THE Knuckey 60-ton, 1000-hp dynamometer truck in Fig. 1 combines three prime functions. It operates as: (1) a truck or self-propelled electric drive vehicle, (2) a dynamometer or electrical power absorption vehicle, and (3) a prime mover or electric-drive towing tractor.

Its basic job is to function as a load vehicle to test the tractive effort of large Army wheeled and tracked combat vehicles. As such, it produces a continuous drawbar pull of 60,000 lb at as low as 2 mph. The vehicle's secondary function is to operate as a prime mover to tow vehicles and measure their tractive rolling resistance. As a prime mover the dynamometer truck rates continuously 30,000-lb tractive effort at 2.5 mph. For self-propulsion purposes the vehicle is capable of 60,000 lb tractive effort and a 45-mph top speed.

Figs. 2 and 3 chart capacities of the vehicle.

Two standard Ford GAN tank engines power the vehicle. They are V-type, 8-cyl, valve-in-head, watercooled engines and are rated 500 hp at 2600 rpm. The engines are directly connected through flexible-disc couplings to General Electric direct-current propulsion generators.

Excitation for the main generator is obtained from one of three amplidyne exciters, which are belt driven by two power take-offs from the engine. The remaining two amplidynes furnish excitation for the two motors. The amplidynes were chosen for the tank job for their extremely fast reaction time and also because the controlling field current could be kept very low. This made it possible to keep the control for the dynamometer truck very small and compact.

During normal truck operations the main generators supply electrical power to the four motors—two for each generator—which are geared to a sprocket which chain-drives the wheels. Also included in the main circuit between the generator and the motors are a bank of load resistors which absorb the load when the vehicle is towed as a dynamometer. These resistors are shorted out of the circuit when the vehicle operates as a truck. The resistors are capable of carrying up to 1800 amp and are called upon to dissipate a maximum of 500 kw continuously.

A fourth small generator, similar in size to the three amplidyne exciters and also belt driven from the engine power take-off, is the battery-charging

* Papers "Sixty Ton Truck Design," by Brumbaugh, "1000-Hp Electric Drive for 60-Ton Field Dynamometer," by Wilson, and "Instrumentation of Field Dynamometer," by Clark, were presented at SAE National Transportation Meeting, Cleveland, March 29, 1949. (Each of these papers is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ each to members, 50¢ each to nonmembers.)

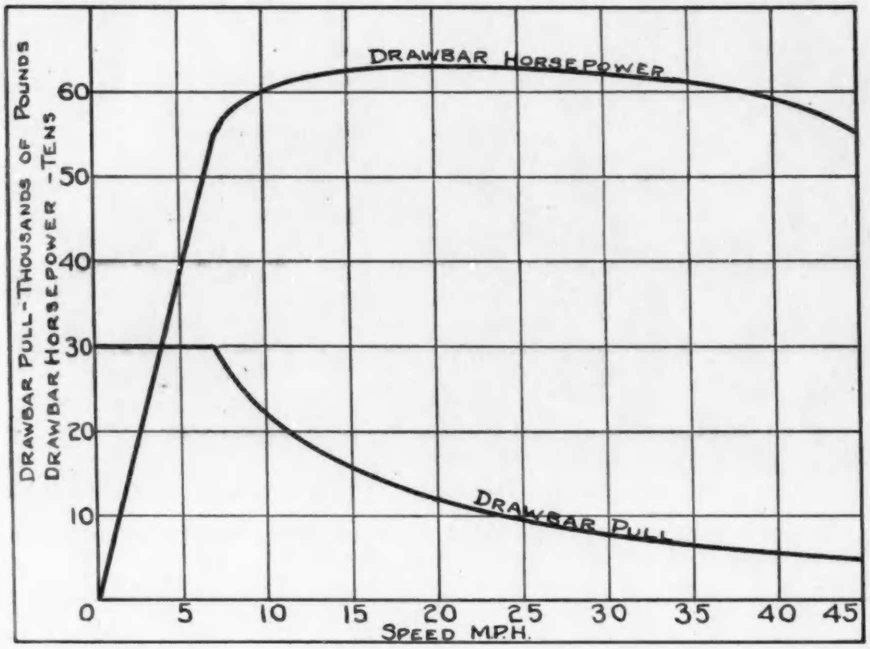

Fig. 2—Minimum full-throttle performance of the dynamometer truck operating as a prime mover

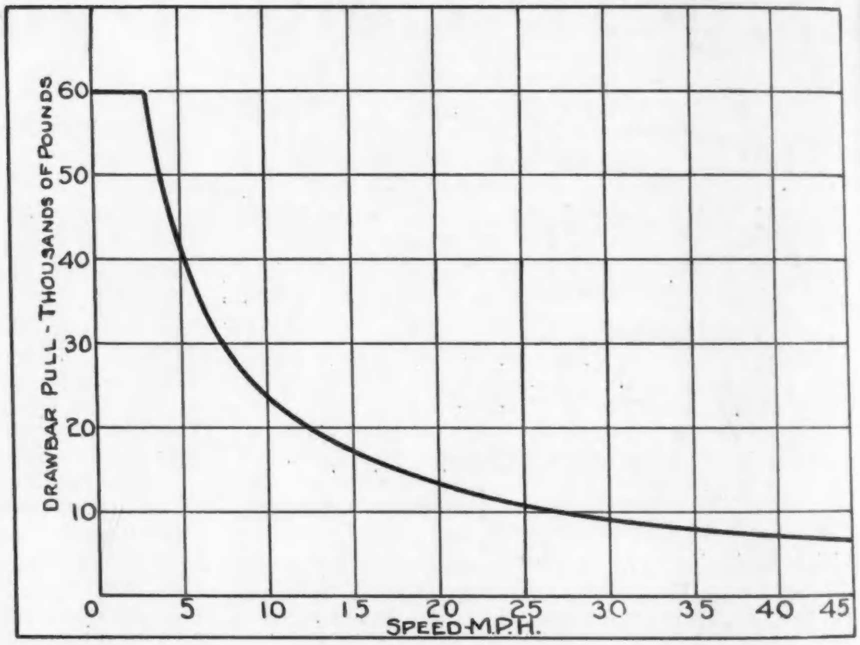

Fig. 3—Minimum power absorbtion capacity of the 60-ton vehicle operating as an absorption dynamometer

generator which furnishes up to 100 amp to the 24-v truck batteries.

The engine cooling system consists of two standard radiators for each engine connected in parallel and mounted directly behind the three amplidynes and the battery-charging generator. The radiators are blown by axial-flow fans mounted on the shafts of the amplidynes and the charging generator.

The propulsion motors are located in pairs symmetrically ahead and behind the bogie centerline, just above the frame rails. The standard tank motors were modified by substituting an end housing and shaft to each, at the pinion end, to take the two-speed section from a heavy truck transmission. The transmission low ratio of 2.3 to 1 allows the motors to turn at reasonable speeds when the dynamometer is operated at very low road speeds and increases the available torque. The transmission is shifted by air cylinders controlled from the cab. Electrical interlocks prevent application of power, unless all four transmissions have completely shifted into the same gear. Since there are no clutches, change of ratio—necessary only for very heavy, slow pulls—is accomplished by letting the truck coast ahead slowly, and then moving the control lever. Colored pilot lights in the cab indicate whether the shift has been completed, and the location of any malfunction.

Prime consideration in design of the frame, Fig. 4, was the provision of adequate strength to resist

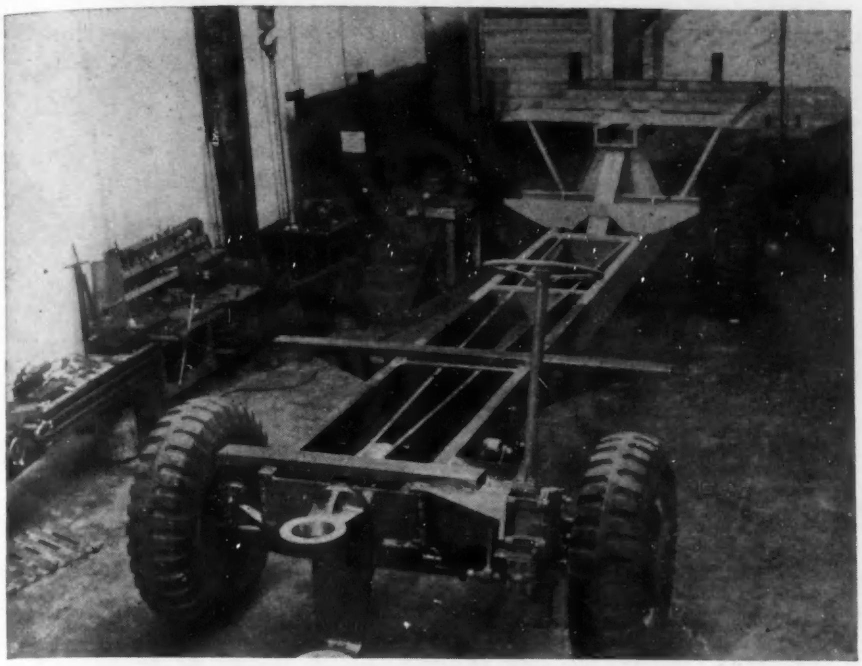

Fig. 4—Structural steel members impart stiffness and rigidity to the dynamometer truck frame

the strains of drawbar pulls up to 150,000 lb, and the shock loads possible when two vehicles of tremendous mass are towing each other. Extreme stiffness and rigidity are necessary to prevent deflection and surges during operation, and weight is advantageous for traction.

Therefore, the frame is fabricated of structural steel with very heavy cross sections. The main rails are welded channels 12 in. deep, with a 3 in. flange and stock thickness of 1 in. These rails are reinforced by a fishbelly section between the front axle and rear bogie, consisting of a 1 in. thick web, with a maximum depth of 8 in. welded to the bottoms of the rails, and a 1×3-in. bar fixed onto their lower edges to form another flange.

The motor supporting structure over the rear bogie forms a truss in conjunction with the main frame rails. This provides the necessary stiffness to support 15,000 lb of ballast hung on the extreme rear of the frame.

Pull from both front and rear drawbars is taken by two additional 1×10-in. longitudinal members arranged roughly in the shape of a diamond, as viewed from the top. Ends of these bars form the mountings for the drawbar brackets, and join the main rails at the bogie centerline; here they are further reinforced by another pair of 1×10-in. bars, approximately 12 ft long, making a solid 3×12-in. section directly over the jackshaft trunnions. This construction affords a straight line pull from towing mechanisms, at both front and rear, to the rear bogie mounting.

Stopping 60 tons of truck from a speed of 45 mph is not easy. In this case the effect of the rotational energy stored in the four propulsion motors and all the heavy drive line-parts had to be considered. The bogie is equipped with 20×8 brakes, and the front axle size is 20×5. It was believed that these would be ample for normal use, but additional stopping capacity would be decidedly advantageous in an emergency.

As originally installed in the tank, each propulsion motor had an internal shoe parking brake of fairly large proportions. Air diaphragms were furnished to permit quick application of these brakes for emergency purposes by means of a trailer hand valve located on the steering column. For parking, a mechanical linkage was provided, operated by a hand wheel in the cab next to the driver.

The hand wheel has been the source of much amusement to the people looking over the dynamometer. But nobody has yet proposed a better way to take up the slack on 40 ft of linkage, including equalizers, for four brakes 15 ft apart, and then exert a pull sufficient to really tie down a vehicle of this size. In normal operation, the air system is used to set the brakes, and the wheel used only as a binder.

Brake System Additions

Layout of the service brake piping is conventional, except that an extra relay is necessary in the line from the application valve to compensate for the long run of tubing required, and a hand valve as well as a foot valve is provided for the service brakes.

The cab of the dynamometer was designed to accommodate in comfort, an operating crew of four men, and most of the instrumentation, as well as the electrical control equipment for the truck itself. Its floor is approximately 9 ft square, and a 6-ft man can stand erect in practically all of it.

The driver occupies a single seat in the left front corner, while the operator who handles the power when testing, and one or two instrument operators sit on a long seat on the right side. Behind them is space for instrumentation equipment and ample room for up to four observers or members of the test crew. Electrical units for truck control are in a large cabinet on the back wall.

The field dynamometer is equipped with instrumentation to measure and record all quantities being tested. The quantities are drawbar pull, ac-

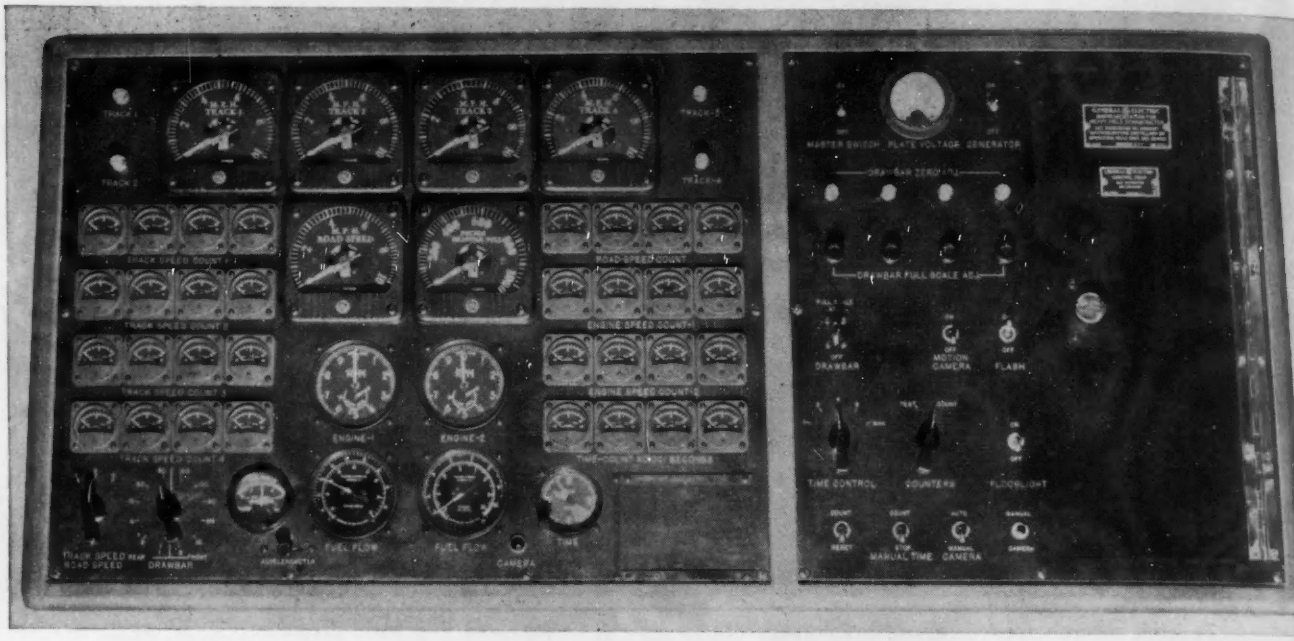

Fig. 5—Instrument and control cabinet of the field dynamometer. The instrument panel (left section) is photographed for recording purposes; the instrumentation is operated from the control panel (right portion)

celeration, road speed, track speed, engine speed, and fuel flow. Instruments and control circuits for the measuring equipment are mounted in a control desk, shown in Fig. 5. This control desk, mounted in the truck cab, is viewed by a camera having a synchronized electro-flash unit. A movie camera is also mounted for recording variations in reading.

Drawbar pull is measured by a calibrated steel section carrying the force, upon which are mounted resistance wire strain gages. These strain gages give an electrical output which is amplified and used to operate an indicating instrument.

To cover the wide range of forces to be measured, both on the front and rear of the truck, six full-scale ranges are provided in each drawbar. This is accomplished by having two strain tubes in series. The low-range tube, having a smaller cross-section, measures forces up to 25,000 lb on the front drawbar after which stops are engaged, protecting this tube against further increase in force. The high-range tube is then used to measure forces up to 150,000 lb. The drawbar is designed to withstand forces up to 400,000 lb without being overstressed. Drying elements are installed to prevent moisture from affecting the resistance wire strain gages.

Since the drawbar forces between the dynamometer and the vehicle attached to the dynamometer can accurately be measured only when acceleration forces are zero, equipment is included for measuring acceleration of the dynamometer. The circuit consists of a d-c tachometer generator, a capacitor differentiating circuit and an indicating instrument. Two scales of 0.2 and 2 ft per sec per sec are provided by means of a shunt on the indicating instrument. In practice, the acceleration of the dynamometer can be held to about 10% of the most sensitive scale or 0.02 ft per sec per sec.

Road speed is measured by a "fifth wheel" of the bicycle type, which drives a generator unit. This unit contains a d-c tachometer, used to operate a conventional d-c instrument, and an a-c generator, is used to obtain an extremely accurate indication of speed by counting cycles of the a-c voltage over a definite interval of time.

Track speed is measured in the same manner as road speed. An identical generator unit is bolted to the sprocket carrying the caterpillar tread. By comparing the calculated speed of the track with the actual speed measured by the fifth wheel, the slip of the track is measured.

A conventional aircraft-type engine-speed indicating system is used for direct engine-speed indication. A counter unit is used to count the cycles from the tachometer generator. Since the frequency of the generator is low (one cycle for two revolutions of the engine), a frequency quadrupling circuit is used to raise the number of counts obtained in a given period and thus improving the accuracy.

An aircraft-type fuel flow meter, based on the rotameter principle, measures fuel flow.

One of the important features of this measuring equipment is the ease and rapidity with which the information on the instrument panel can be recorded. Two cameras turn the trick.

The first camera is a 35 mm Kodak "35" camera with a special film-winding mechanism. A synchronized electronic flash unit provides light for the photograph independent of the outside lighting conditions, other than direct sunlight on the instrument panel. When test conditions have stabilized, a push button starts the speed counters; they operate for a preset interval. At the end of this interval, a 1-sec delay allows the instrument pointers to come to rest; then the shutter of the camera and the flash unit are automatically operated. The film winding and shutter reset mechanism operates and the equipment is ready for the next recording in approximately 4 sec.

The second camera is a standard 35 mm movie camera using photoflood illumination. This camera records variations in the indications during special tests.

Automotive CORROSION CLINIC

Rx

BASED ON PAPERS* BY

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H. A. Pray

BATTELLE MEMORIAL INSTITUTE

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Cleveland Research Division,
Aluminum Research Laboratories,
ALUMINUM CO. OF AMERICA

Body Steel Corrosion[1]

Cause:

Most serious car body corrosion proceeds from underneath or inside the exterior surfaces. Reason: these surfaces get less attention from the owner. They are kept wet longer by splash and clinging mud, which retain and concentrate corrosive substances. Salt on roads and steel composition also influence corrosion rate of body steels.

Considerable confusion exists on effect of sodium or calcium chloride sprinkled on roads to melt ice or lay dust. For example, tests with sodium chloride solutions—ranging from pure water to brine—yield greatest corrosion of bare steel with very small salt percentages. And pure water is more corrosive than all salt concentrations above 1 or 2%.

Explanation for this is that solubility of oxygen in sodium chloride solutions decreases as the salt concentration increases. Lack of oxygen in more concentrated solutions more than compensates for corrosion-aggravating effect of the chloride ion.

This reveals why cars, driven in areas where roads are never salt treated, corrode almost as much as those in regions treated regularly.

Apparently humidity of the atmosphere where the car is kept counts more than salt on roads in determining extent of underbody corrosion. But salt adhering to the car may have an indirect adverse affect by keeping the metal moist through its hygroscopic properties.

Steel composition also can accelerate body corrosion. Rimming steels, used because of forming and drawing characteristics, may contain an unfavorably high ratio of sulfur to copper or nickel content. This lowers the steel's atmospheric corrosion resistance.

Additionally, certain components not requiring

* Papers "Corrosion Problems of the Automotive Engineer," by LaQue and Hergenroether, "Corrosion of Electroplated Steel in Automotive Applications," by Pray, and "Service Tests Solve Aluminum Cylinder Head Corrosion Problems," by Daugherty and Koenig, were presented at SAE Summer Meeting, French Lick, June 7, 1949. (Each paper is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ each to members, 50¢ each to nonmembers.)

[1] From paper by LaQue and Hergenroether.

severe forming or drawing operations may be made of commercial quality rather than deep-drawing quality sheets. Chance of finding an unfavorable composition from the corrosion standpoint is greater in these parts. Perhaps this explains why parts in this class are particularly susceptible to corrosion.

Electrical currents for lighting, carried by the car body, also may accelerate corrosion. Trouble from this source stems from an interrupted electrical connection between two adjoining surfaces. Also, if the joint has a high enough resistance to force the current to leave the metal and flow around the joint through a moisture film, corrosion will set in on the side where the current leaves the metal. It might blister the paint by forming alkali on the other side.

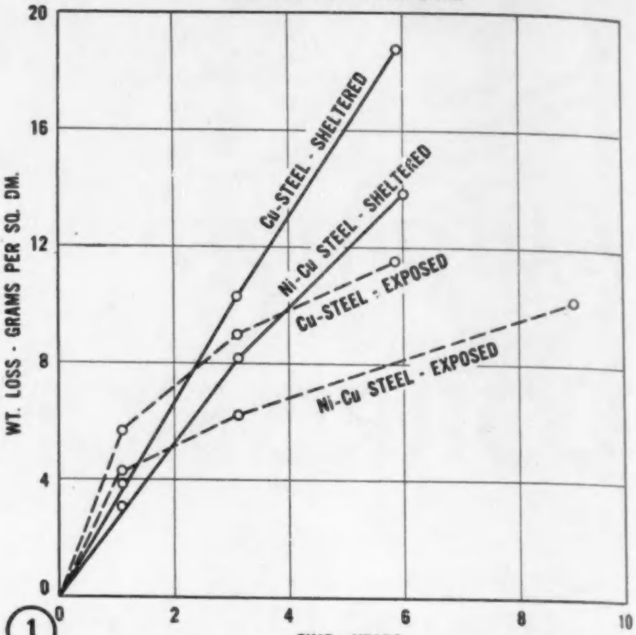
Remedy:

Quick and thorough drying is the most direct way of nipping underbody corrosion in the bud. It also affects the protective value of steel rust films.

Tests emphasize this point. Specimens exposed in the shade under a partial shelter corrode more than those boldly exposed to rain and sun. This difference increases with duration of exposure, reflecting the greater protective value of rust that has a chance to dry. Fig. 1 illustrates this.

Car underbodies should be designed with quick drying in mind. Such controlled underbody ventilation could include air channels and guiding vanes to direct warm drying air from under the hood to surfaces in need of the drying effect. Also effective are fiber-reinforced asphaltic coatings on underbodies, if applied properly.

EFFECT OF SHELTER ON SPECIMENS EXPOSED VERTICALLY AT BAYONNE



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Efforts are being made to relieve the possible corrosive effects of salts used for road treatment by adding corrosion inhibitors to them, especially polyphosphates and chromates. But car users can help themselves most by keeping or parking their cars where they will dry quickly, and by washing them often enough to get rid of accretions that may

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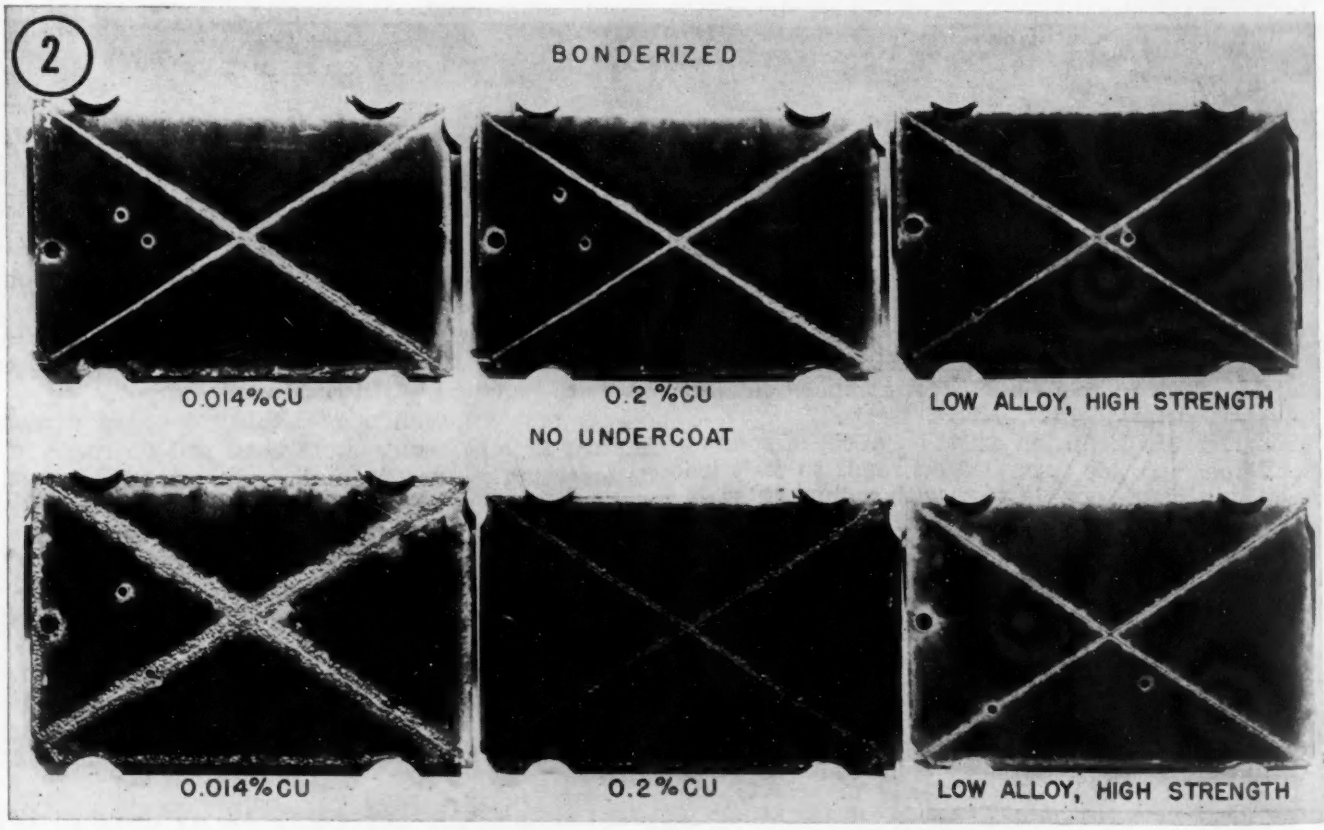


accelerate corrosion. Danger of accelerated corrosion from salts is not great enough to recommend discontinuance of this accident-preventive treatment.

On the steel composition side, corrosion resistance could be improved by using small amounts of copper to overcome sulfur's adverse effects. Desirability of using steel with a 0.05% minimum copper content was confirmed by a survey which showed that 90% of specimens of severely corroded car body parts contained less than this amount of copper. The investigation also disclosed that corrosion at its worst came from a low copper content combined with high sulfur content in parts not severely formed, but used in corrosive localities—such as rocker panels.

A 0.05% copper content provides a barely satisfactory corrosion resistance level. For corrosive environments, sulfur content of steel containing the minimum copper should be held under 0.04%.

Steel's corrosion resistance improves by raising the copper content to about 0.25%, as in ordinary copper steel, and by adding larger amounts of nickel or chromium, as in common low-alloy high-strength steels. Combinations of alloys—copper and nickel—in amounts not much greater than current residual percentages might be effective. It could be done with little additional cost and without sacrificing working characteristics. Such steels are under investigation, with preliminary results encouraging.

Even under the mask of a body finish the corrosion resistance of the steel base shows up. In tests now underway in the marine atmosphere at Kure Beach, N. C., a typical body finish was applied to steels of different composition, ranging from a very low copper content to two typical low-alloy high-strength steels. Half the specimens were given a commercial bonderizing pretreatment before painting; the balance were coated with a primer and top coats without the phosphate undercoat.

Fig. 2 shows effects of both steel composition and phosphate treatment. It illustrates the conditions of the scribed specimens after exposure of about 2 months, 80 ft from the ocean. These results point up the protective values of desirable steel composition and bonderizing. Probably optimum results stem from use of phosphate treatment on a mildly alloyed steel.

Car body corrosion from stray currents can be remedied by providing a metallic electrical bond around any insulated joints.

Corrosion of Electroplated Steel[2]

Cause:

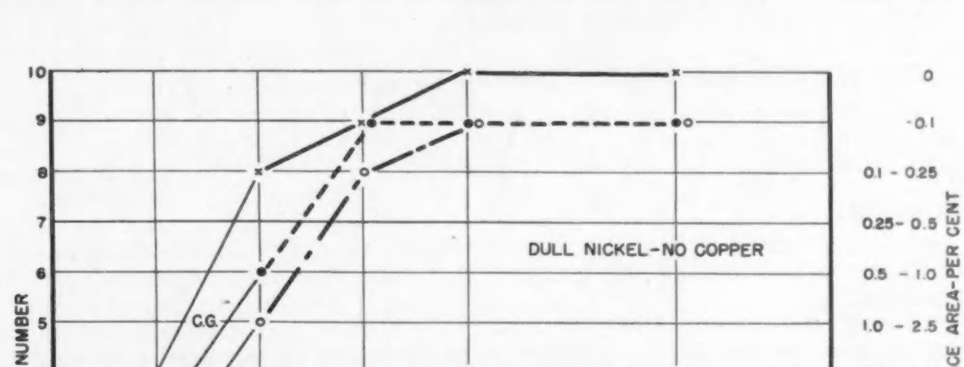
Atmospheric corrosion tests of plated steel panels at six different locations put the finger on both atmospheric severity and plate thickness as the chief factors in plating life.

These panels were exposed to the weather at (1) New York (industrial), (2) Kure Beach (seacoast), (3) Pittsburgh (heavy industrial), (4) State College, Pa., (rural), (5) Coral Gables (semitropical, inland, suburban), and (6) Detroit (where the industrial atmosphere was supplemented by thrice-weekly salt solution applications to simulate winter driving conditions).

Ratings after a year and one-half exposure at five different locations are shown plotted as a function of nickel thickness in Fig. 3. These are for specimens with nickel plated directly on high-carbon steel, with no copper undercoat. The chart shows wide differences in effect of atmospheric conditions on such coatings.

Pittsburgh's heavy industrial atmosphere is most damaging, with New York less severe, but more corrosive than the Kure Beach seacost environment. Less contaminated atmospheres of Coral Gables and State College produced milder corrosion. Essentially the same effects were noted for the four nickel plate thicknesses used in the exposure tests at these locations.

Using copper in place of any appreciable amount of nickel speeds up deterioration, the tests also



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[2]From paper by Pray.

show. The higher the proportion of nickel in composite plates of a given total thickness, the better its corrosion resistance. Buffing the copper deposit prior to nickel plating may be slightly beneficial in the early weathering stages, but has no important effect on the shape of deterioration-time curves.

Remedy:

These tests show clearly that nickel thickness is the biggest single element that determines useful decorative life of such coatings at all the exposure stations. The other variables—type of steel, copper undercoat thickness, buffed or unbuffed undercoat, and even type of nickel plate—are only of secondary importance.

The indication is to plate steel bright work of cars with as liberal a thickness of nickel, either directly on the steel or over a copper undercoat, as the economics of the situation permits. Of course the decorative life of any nickel-chromium or copper-nickel-chromium electroplate on steel depends on how and where the coating is used. Over this factor the car producer has no control.

Engine Cylinder Corrosion[3]

Cause:

Much of engine cylinder deterioration labelled wear includes a large corrosion factor.

Corrosive influences include acids originating in sulfur compounds in fuel, oil, or air; bromine compounds from antiknock fluids; and such corrosive agents as acetic and formic acids formed by incomplete combustion. But there is much more of ordinary carbon dioxide, a principal combustion product, than the other substances. It is sufficiently corrosive to account for the damage experienced. The other agents may mainly influence the action of carbonic acid.

Most engineers tend to underestimate the corrosive power of carbon dioxide in solution. An investigation of corrosion by water saturated with various carbon dioxide and air mixtures at temperatures from 160 to 250 F, showed:

1. Initial corrosion rates of iron may reach 0.33 in penetration per year.
2. Corrosion can proceed rapidly even in the absence of oxygen.
3. Corrosion rates decrease with time, evidently due to formation of adherent protective corrosion products.

With regard to the last item, it seems likely that continual abrasion of the cylinder surfaces by piston ring travel will remove protective corrosion products as fast as they form. Consequently the high rates of attack observed in short test periods probably will prevail.

Additionally, presence of other compounds in the corrosive condensate—such as sulfurous and organic acids—may serve chemically to destroy protective carbonate corrosion products, accelerating corrosion in this indirect way.

Cylinder surface temperatures below the dew point of combustion gas moisture induce wear. This accounts for the greater corrosive wear in engines that start and stop frequently—especially in cold weather. It also explains the benefits from cooling system adjustments to keep upper cylinder walls (where corrosive compounds occur) above the dew point, even under adverse operating conditions.

Other factors that influence corrosive wear include the lubrication system, protective qualities of the lubricant, and piston rings functioning. Nature of the fuel and richness of the fuel-air mixture probably affect the way the fuel burns, and determine both operating temperatures of cylinder walls and how long they remain at a corrosive temperature.

Remedy:

A most effective way of reducing corrosive wear is to improve corrosion resistance of the cylinder material.

Adding small amounts of nickel, chromium, copper, and molybdenum to cylinder irons produces moderate improvements. Big gains come from switching to the many times more resistant austenitics cast irons, of which Ni-Resist is the most common. Such corrosion-resisting austenitic irons could be used as inserts for those cylinder parts near the top of piston travel, where corrosive forces prevail.

Table 1 shows the relative performance of Ni-Resist and cast iron in corrosive media believed to corrode cylinders.

Corrosion-resistant materials for cylinders also must provide desirable frictional properties. This is no barrier to use of austenitic cast irons. They benefit from both a graphitic structure, similar to cast iron, and work-hardening characteristics of the austenitic matrix; it makes possible surfaces with excellent mechanical as well as corrosive-wear resistance.

This combination of properties shows up in excellent service records. They are most striking in service with frequent starts and stops and which extends to sustained heavy-load conditions, as in long-haul trucking. Table 2 gives typical service records.

[3] From paper by LaQue and Hergenroether.

Chrome plating advantages in extending cylinder life come largely from the added corrosion resistance as well as extra hardness and good frictional properties. Corrosive agents, that may penetrate chrome plating and undermine it by corroding the iron beneath, may be more important than mechanical factors in determining useful life of chrome plated cylinders.

Table 1—Comparative Resistance of Ni-Resist and Cast Iron to Corrosion by Chemicals Encountered in Exhaust Condensate

Corrosive Medium	Temperature	Duration of Test	Rates of Corrosion in inches per year: Ni-Resist	Rates of Corrosion in inches per year: Cast Iron
Mixed Acetic and Formic Acid Vapors and Steam	212–220 F.	150 hr	0.026	0.142
Water Saturated with Carbon Dioxide	60 F.	....	0.0017	0.027
Sulfuric Acid 0.1%	200 F.	166 days	0.023	0.067
Sulfuric Acid 5%	60 F.	....	0.006	1.23
Hydrochloric Acid 1.8%	60 F.	....	0.012	1.07
Wet Exhaust Fumes & Condensate from Gasoline Engines	....	99 days	0.012	....

Table 2—Comparative Service Records of Ni-Resist and Other Cylinder Materials

Service	Average Cylinder Wear—Inches: Ni-Resist	Cast Iron	Chrome-Plated Cast Iron	Water Side Corrosion: Ni-Resist	Cast Iron
1. Diesel engine in heavy-duty truck hauling ore from pit	0.0036[a]	.....	0.0027[a]	None	Occasional Perforation
2. Diesel engine in heavy-duty truck in West Coast road service	0.00015[b]	0.0015[b]	.....	Slight Pitting	Occasional Perforation
3. Taxicab and light city bus service with Ni-Resist in ser in grey iron block	1[c]	19[c]	.....	......	....
4. Light-duty high-speed aircooled engine with Ni-Resist liner in aluminum block	0.00025[a]	.....	.....	......	....

[a] per 1000 hr [b] per 10,000 miles [c] relative values

Cylinder Liner Corrosion[4]

Cause:

Corrosion of the water side of cylinder liners may become serious in diesel engines, although ordinarily it is not a problem in gasoline engines. Most troublesome corrosion of this sort is highly localized and looks like that in Fig. 4. (The photograph at left represents cast iron after vibratory cavitation erosion test in salt water; the one at right is a cast-iron diesel cylinder after service with salt water as the cooling medium.)

Similarity of this damage to that produced in accelerated cavitation erosion tests by the vibratory method suggests this theory: Vibration of the cylinder walls in contact with the cooling water may cause this damage—just as vibration of laboratory cavitation testing apparatus produces damage practically identical to that on cylinder liners.

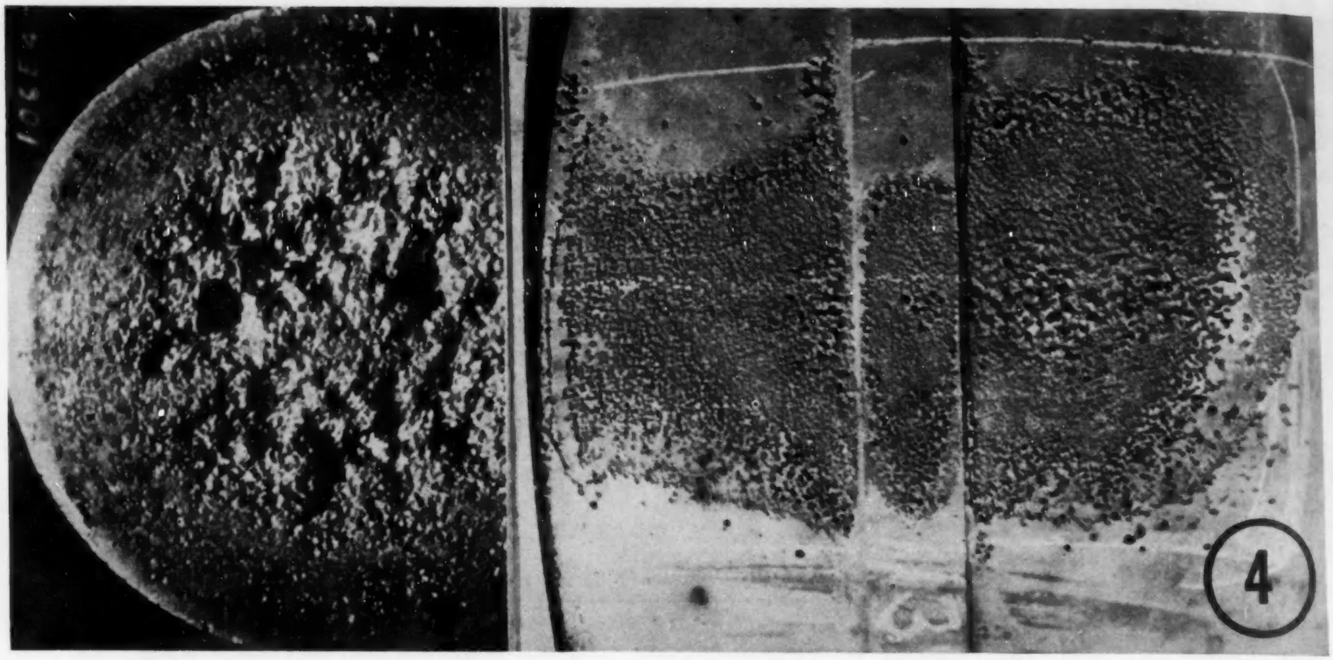
[4] From paper by LaQue and Hergenroether.

In addition to the cavitation erosion type of deterioration, cylinder liners occasionally are subjected to severe localized corrosion within any crevices that may exist in water spaces. Such crevices may form where two pieces of metal rest on each other—like supporting ribs between jackets and cylinders. Mechanism of this damage involves an oxygen concentration cell, or differential aeration cell, in which the surfaces with the crevice that are shielded from free access to oxygen becomes anodic to surrounding freely exposed metal.

Remedy:

If vibratory cavitation erosion actually is a major factor in this peculiar cylinder liner deterioration, then these steps should be helpful in treating it:

1. Check the mechanical design to minimize vi-



bration by stiffening or supporting other members that will dampen out harmful vibrations.

2. Keep cooling water temperature and pressure at as high a level as practical.

3. Introduce air bubbles into the cooling system to cushion the cavitation blows.

4. Reduce or eliminate the corrosion phase of cavitation erosion by (a) using more corrosion-resistant liners and (b) using plenty of chromate inhibitor.

Chromate inhibitors prove quite effective in preventing corrosion or erosion of this type. The concentration required is from 2000 to 5000 parts per million, with the water usually being adjusted to a pH between 8 and 9. Some inhibitors are effective at a pH of 7, considered more desirable than the higher pH with aluminum in the system.

Use of chromates must start as soon as the engine is put into service and proper concentration maintained at all times. Once deep pits or fissures pit the iron surface, effective corrosion inhibition at bases of such pits and fissures becomes difficult.

These usual passivating inhibitors cannot stop severe localized corrosion within crevices, since they cannot reach the surfaces requiring protection. They may even promote corrosion by increasing the potential difference between the shielded and freely exposed surfaces. Best solution to this problem is either to eliminate such crevices if possible, or to use caulking compounds that seal them thoroughly. Protective coatings also might be applied in the critical areas.

Muffler and Tail Pipe Corrosion[5]

Cause:

Corrosion inside mufflers and tail pipes come from the same causes that make for corrosive wear of upper cylinder surfaces. Chief differences are:

1. Muffler or tail pipe temperatures will be lower than in a cylinder so that condensation will occur more often and for longer periods.

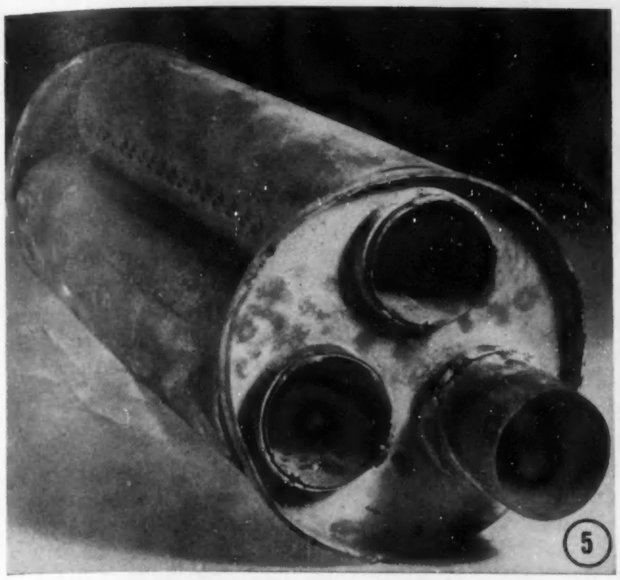
2. More opportunity exists for corrosive condensates to collect or accumulate in a muffler.

3. Since erosive forces in the muffler are less severe, protective corrosion-product films may have a better chance to form and to last.

Mufflers corrode severely and rapidly most often in cars run long enough to form corrosive condensates, but run too short for above-dew point temperatures to evaporate them. Cars used by doctors, public utilities, and those used for short shopping trips induce such conditions. In normal car use, temperatures high enough to destroy muffler or tail pipes by oxidation or scaling occur infrequently.

Corrosion from the outside as well as from within attacks mufflers and tail pipes. Car use will determine which effect will predominate. Operation keeping both inner and outer surfaces dry will minimize corrosion. Where temperatures are high enough to keep inner surfaces dry, but outer surfaces merely warm and not dry, the higher temperature will accelerate outside corrosion. Where both

[5] From paper by LaQue and Hergenroether.



inner and outer surfaces are wet often, two-directional attack of the steel will minimize its life.

Remedy:

The engineer has less opportunity to prevent condensation in mufflers than he has to prevent condensation in cylinders since conditions of use chiefly determine muffler operating temperature. But he should direct his design toward elimination of condensates.

The design and construction provide minimum opportunity for condensate to collect, maximum opportunity for its removal by both evaporation and entrainment in the exhaust gas stream.

From the materials and coating approach, several possibilities present themselves.

First, highly corrosion-resistant materials, such as Inconel and stainless steel, will eliminate muffler and tail pipe corrosion problems. For example, Fig. 5 shows the Inconel interior parts of a bus muffler after 179,000 miles of service, and they were as good as new. During that time it had three outer shells—one of steel and two of aluminum-coated steel.

Doubling steel thickness will double the life of ordinary steel mufflers. This procedure has the advantage of introducing no new fabrication or joining problems. And steels alloyed with small amounts of nickel, chromium, and copper are superior to carbon steels.

Steel coated with zinc will last a little longer, depending on the zinc thickness. Same is true of lead-tin alloy coatings. Steel hot-dipped in aluminum or aluminized is about twice as durable as bare steel. A nickel-zinc coating—corronizing—approaches the aluminized treatment. While vitreous enamel coatings are highly protective, their application calls for special care in muffler design and fabrication.

Aluminum Cylinder Head Corrosion

Cause:[6]

Available evidence shows aluminum cylinder head corrosion to be galvanic. It is found in a severe form adjacent to junction lines with iron and appears independent of port size or contour.

For example, severe aluminum head corrosion has been noted at water passages in the aluminum cylinder head at point of contact with block and gasket. Also substantiating the galvanic theory is the fact that aluminum cylinder heads that corroded severely on iron blocks, do not corrode appreciably on all-aluminum blocks under similar conditions.

Leakage of coolant together with galvanic corrosion products into stud clearances make aluminum cylinder heads stick to the studs.

Laboratory galvanic studies with electrically coupled iron and aluminum specimens immersed in various coolants, show some reasons why inhibitors that protect iron from corrosion may not be effective for aluminum.

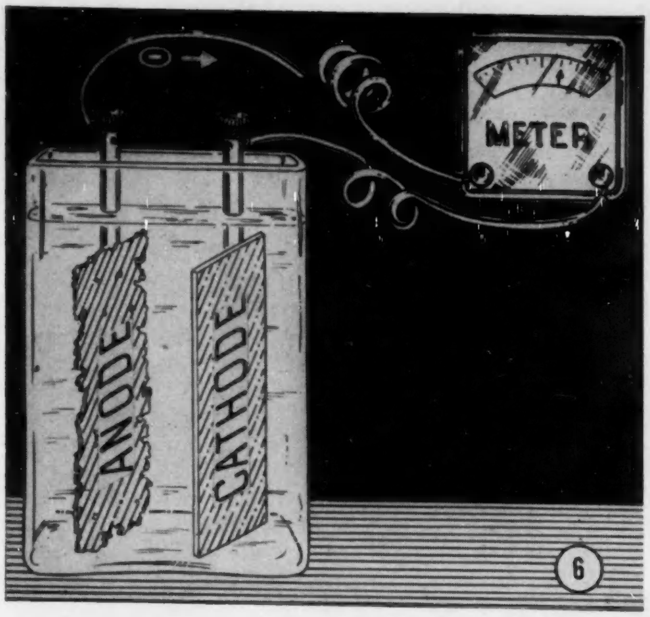
In such a galvanic cell, Fig. 6, electrons flow in the external circuit from the anode, or corroding member, to the cathode, or noncorroding member, as indicated by the milliammeter. Direction of electron flow, caused by higher potential of the anode and magnitude of the measured current, show directly the rate of galvanic attack on the anode. Factors such as metal composition, liquid in contact with the couple, temperature, polarization, aeration, and metal surface films influence variations in electrode potentials of the aluminum-iron couple.

Tests with prepared iron-aluminum couples in distilled water and high-chloride water, both uninhibited, showed iron to be the anodic or corroding member and aluminum the cathodic or noncorroding member, with electron flow from the iron to the aluminum. Iron rather than the aluminum corroded galvanically, despite the penetrating action of chlorides on the aluminum oxide film. Apparently the corroding iron protects the aluminum from galvanic attack in these uninhibited liquids. This process of corrosion protection at the expense of another metal is called "cathodic protection."

In another test the iron-aluminum couple was immersed in a solution of high-chloride water and a glycol antifreeze, containing an effective soluble chemical type of iron inhibitor. But now the aluminum becomes anodic and suffers appreciable galvanic attack because the inhibitor lowers potential of the iron below that of the aluminum. Electrons have reversed their flow, passing from the corroding aluminum anode to the noncorroding iron cathode.

In dealing with aluminum cylinder head corrosion, one must not overlook the serious corrosion problem

[6] From discussion by D. H. Green, National Carbon Co.



involving iron, copper, brass, and solder parts of the conventional automotive cooling system, as well as effective control of this corrosion with proprietary antifreeze inhibitors. Importance of corrosion control of iron and other commonly-used cooling system metals cannot be over-emphasized.

Remedy:[7]

The engine manufacturer as well as the vehicle operator can take steps to eliminate aluminum cylinder head corrosion. And most of the measures entail little if any added cost.

For example, the aluminum alloy used in the cylinder head should contain sufficient copper (at least 3%) and a low enough concentration of zinc (0.4% maximum) to provide a satisfactory low electrode potential. Low potential also stems from using the cylinder head in the as-cast or solution heat-treated condition. Because of its lower cost, the as-cast condition is preferred.

Cylinder head gasket design bears heavily on port corrosion of aluminum cylinder heads. In conventional design, the hole in the gasket frequently is larger than the corresponding port in the head so that an aluminum shoulder is exposed to coolant flow. Port corrosion generally starts on these shoulders.

A gasket with ferrules that protrude into the cylinder head ports will substantially prevent corrosion under otherwise corrosive conditions, cyclic engine tests have shown. Such a gasket, as in Fig. 7, might be too expensive for car cylinder heads, but could prove valuable where cost is less important.

The material used for the ferrules or grommets of the protective gaskets should resist corrosion and erosion. And it should have relatively little tendency to cause galvanic corrosion of the aluminum cylinder head since considerable area of such ferrules would be exposed to the coolant.

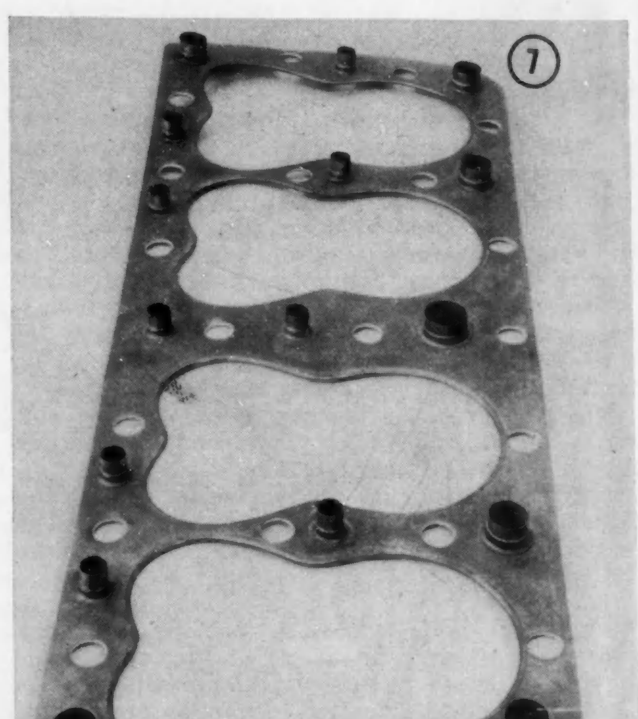
That these requirements are met by 8-lb terneplate, 18-8 stainless steel, or lead was indicated by cyclic engine tests with a standard corrosive coolant and by electrode potential measurements in salt-peroxide solution. Copper also was found satisfactory in a one-year cyclic engine test of the ferruled gasket in Fig. 7, despite the comparatively low electrode potential of copper.

Almost equally effective protection against corrosion can be achieved with a flat protective gasket, at no higher cost than conventional gaskets. Making gasket holes only 1/8 in. smaller in diameter than the corresponding cylinder head ports gave satisfactory protection against corrosive coolants in cyclic engine tests. Where head and block ports differ in diameter and no constriction at the gasket hole can be tolerated, the gasket port should equal the head or block port diameter, whichever is smaller.

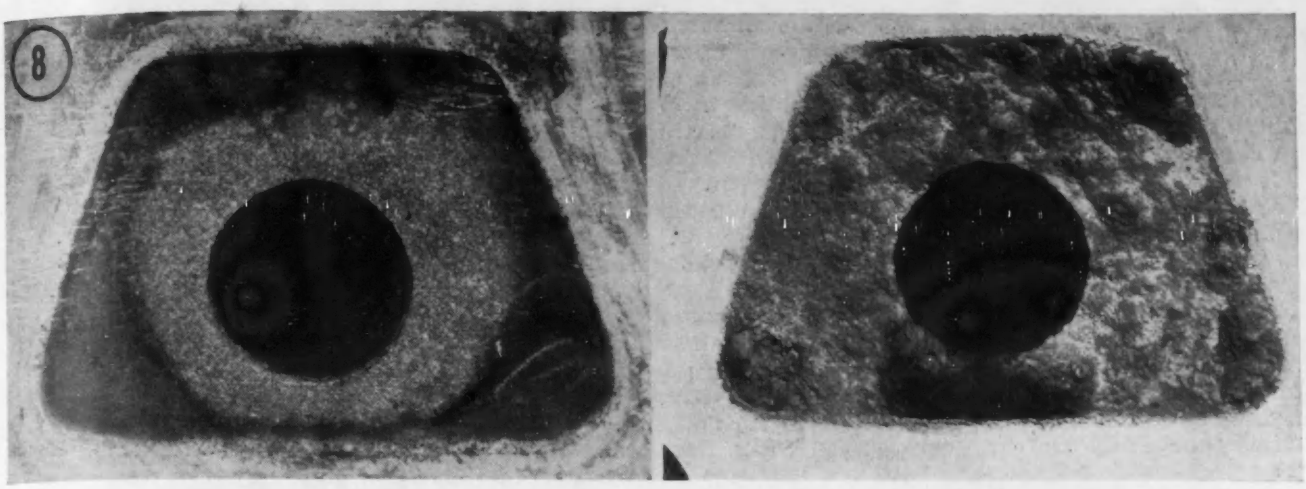
The material from which cylinder head gaskets of conventional design are made appears to be only a minor factor in aluminum cylinder head corrosion. This is probably because the area of gasket exposed to the coolant is too small to affect substantially the galvanic process. Such relatively small cathodic areas become polarized readily. Nevertheless, steel-asbestos type gaskets have been considered preferable to copper-asbestos to achieve some reduction in galvanic currents.

Cylinder block materials have not been studied; but it would help if methods were available to alter their solution potentials or polarization characteristics.

A naphthenic base soluble oil is recommended



[7] From paper by Daugherty and Koenig.



as an inhibitor for water to reduce cylinder block rusting and resultant radiator clogging. The oil should consist of a mineral lubricating oil, emulsified mainly with naphthenic acid soaps. The cooling system should be clean before the soluble oil is introduced. The oil should be pre-emulsified with at least one and one-half parts by volume of water by pouring the oil into the water and agitating vigorously. The cooling system should be flushed twice yearly and fresh soluble oil added.

Chromates or dichromates never should be used with engines having iron blocks and aluminum heads; severe corrosion of aluminum will result. This is demonstrated in Fig. 8. Both these aluminum heads were operated 12 weeks in cyclic tests with cooling water containing 273 ppm of chloride. The one at left ran without inhibitor; the one at right contained 3 g per liter of sodium dichromate. The results speak for themselves.

However, these compounds may be useful in cooling systems where no conditions conducive to galvanic corrosion are present.

Antifreezes recommended for general use with engines having iron blocks and aluminum cylinder heads is a solution of any reputable proprietary ethylene glycol antifreeze in water; it should contain at least 50% by volume of ethylene glycol. More dilute glycol solutions and alcohol antifreezes can be used with the recommended alloy and, particularly, with a protective gasket design. Compounds and "neutralizers" used in cleaning cooling systems should be thoroughly flushed out after use. Since chloride accelerates corrosive action of antifreezes and inhibitors, all practical precautions should be taken to keep the chloride concentration of coolants at a minimum.

Use of kerosene as a coolant would eliminate all cooling system corrosion problems. Kerosene may be a satisfactory coolant for some applications. Engine and radiator design could be altered to accommodate its heat capacity and conductivity characteristics. But after such alterations, the engine probably would not operate efficiently with aqueous coolants.

Hose lined with certain synthetic rubbers will resist deterioration by kerosene. Thermostatic signals can be used to warn against an excessive rise in temperature. Special processing at a practical cost can reduce odor and volatility. Additives to kerosene are available to control leakage and accidental contaminations with water.

However, kerosene may introduce a fire hazard with conventional cooling system designs.

Electrolytic protection of aluminum cylinder heads by galvanic anodes or battery current is impractical. Paint or chemical coatings to protect aluminum cylinder heads have so far proved unsuccessful, except in engines with top coolant temperatures of 150F.

Many of these recommendations could be ignored if an aluminum cylinder block were used with an aluminum head.

Any one of the following four methods can effectively eliminate the difficulty in removing cylinder heads:

1. Using cap screws instead of cylinder head screws.

2. Making stud holes 1/16 in. greater in diameter than the studs, instead of the usual 1/32 in.

3. Coating the studs with extremely heavy petroleum bright stock having a viscosity at 210 F of 300 SUS or more, and a flash point of 625 F or higher.

4. Using inserts in the stud clearances.

The first two methods should be adopted by the engine manufacturer. The petroleum coating is recommended for new engines where small stud clearances must be used, and for replacing aluminum cylinder heads with small clearances. Inserts are effective, but relatively costly.

The recommended diameter clearance of 1/16 in. has been used with aluminum V-8 cylinder heads in Canada for the past 15 years. It is reported that no sticking problem has been encountered. The Canadian practice is to locate the head by providing only a few thousandths of an inch clearance at one of the central studs in the bottom row, flattening the end holes in the top row to restrict vertical play to a few thousandths. Other methods of location would be the use of dowels or various arrangements of flattened holes.

DESIGN Auxiliary

BASED ON PAPER* BY

Ralph K. Super

Timkin-Detroit Axle Co.

STOPPING the vehicle and holding it within reasonable operational limits form the real criteria of auxiliary brake acceptability. Design limitations of most auxiliary brake types and locations make these seemingly simple requisites tough to meet.

The auxiliary brake must fulfill two objectives: (1) sufficient capacity and power to bring the vehicle to a safe stop and (2) ability to hold the vehicle at rest.

Setting up specific distances or brake performance on stopping ability without due regard to loads being handled constitutes a hazard. In many cases, a bus or truck auxiliary brake powerful enough to stop the vehicle in less than 40 ft from 20 mph also can impose large enough torque loads on drive line, gear set, axle shafts, or wheel drives to make these parts fail. (Statutes require a 50-ft stop from 20 mph.)

So-called emergency brakes often induce failure of connecting parts in uncontrolled application, as in an actual emergency. Such brakes contribute little to safety. Springs were installed in pull rods of many of these brakes to limit maximum effort that could be imposed. This type of brake is not the ultimate in design, nor is a specified stopping distance the most desirable basis for evaluating brake performance.

Major consideration here is the ability to make a safe, rather than a violent or dangerous quick stop. Need for an auxiliary brake able to make a fast stop under emergency conditions is impractical in present-day traffic. The operator's mental and physical reaction time to a service brake system failure leaves very little time for effective use of a secondary brake system. In most cases attention to steering offers the better alternative.

Best method for determining auxiliary brake adequacy appears to be grade ability. This has practical value from a safety viewpoint since any driver wants to be able to stop the vehicle in case of service brake system failure. Particularly is this true with the vehicle on a down grade. Here the auxiliary brake should be capable of stopping the vehicle. Such basis for evaluating auxiliary brake ability recognizes various drive system limitations when the brake is mounted in this system. It also recognizes power limitations and greater capacity of the auxiliary brake mounted in the rear wheel location.

The holding function, second auxiliary brake objective, implies more than most suspect. At first it seems that a locking ability would suffice. But it does not cover many year-'round driving conditions for commercial vehicles or passenger cars. Many have experienced trying to climb a snow or ice-covered grade and found that driving wheels have reached the limit of their tractive ability. Service brake application introduced tractive ability of all the wheels and made it possible to keep the vehicle from sliding down grade.

Point of this illustration is that if the auxiliary brake is set on the vehicle, it invariably operates on the drive wheels only and tractive ability is no greater than when driving. Result: the vehicle slides off the road. This inability to use fully tractive ability of all the wheels on the ground through the auxiliary brake system leaves much to be desired. Such condition is extremely serious with vehicles in combination, where only one or two axles of five or seven are connected to the auxiliary brake.

Only effective practical solution to this problem has been the introduction of actuating means that permit application of service brakes at the same time the auxiliary brake is set. This insures full use of tractive ability of all wheels on the ground for holding purposes. Many truck-trailer combinations have independent power control by which the brake can be set to do this. It generally takes the form of a hand control valve operating the entire service brake system.

Best arrangement to date on commercial vehicles with this equipment is to have the auxiliary brake lever manually operate the auxiliary brake and simultaneously pick up the operating valve of the service brake system. This means the two systems are interconnected for holding the vehicle.

Despite its considerable merit, this arrangement runs the danger of careless maintenance of the auxiliary brake system. Gradual loss of holding power in the service brake system through leakage may eventually impose the complete holding load on the auxiliary brake. If efficiency of the auxiliary brake is not maintained at a high level, it will not hold the vehicle as it would have to if operating as an independent mechanism.

Probably every type of band, shoe, or disc brake has been tried as an auxiliary brake because of its low cost or good performance; but almost all have drawbacks.

For example, the external contracting band brake, a low-cost installation with good stopping ability, has been widely used; but it suffered from abbreviated life in most cases. The brake drum diameters could be held reasonably small and rotative unbalance was not too difficult to hold to a minimum, even under abusive service.

Using thicker linings to lengthen the life of this brake design is the refinement found in the external contracting shoe brake. But both these types have

* Paper "Auxiliary Brakes for Automotive Vehicles—Problems and Solutions," was presented at SAE Metropolitan Section, New York, May 19, 1949. (This paper is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

Problems Deter Brake Aims

poor cooling characteristics since the outside bands or shoes almost completely blanket the hot skin surface of the brake drum at the point of maximum air flow. The band brake's construction simplicity and resultant low cost and light weight make for its wide use as a "parking brake" type of auxiliary brake. This is understandable in the light of its limited heat-dissipating capacity.

The internal expanding shoe brake makes it possible to improve this feature; it permits an externally-ribbed brake drum that increases the cooling surface. This design requires a larger diameter brake drum, and poses greater dynamic balance problems than the previous type.

The internal expanding brake also has the undesirable characteristic of "wash out" due to drum expansion under heat; this requires additional shoe travel. But its greater cooling surface minimizes this feature. The design is widely used as the "emergency brake" type of auxiliary brake. Ultimate in this design is the wheel brake on the rear axle. Used as both a service and auxiliary brake, it offers the brake size and capacity highly desirable in the auxiliary brake system.

Currently-used special designs have such high capacity features. One is a ventilated disc type and the other a drum type. Both designs clamp the drum between an arrangement of inner and outer brake shoes. Both also feature exposed drum areas for cooling. Exposed area amounts to 75% of the total. The disc-type brake is known commercially as the Tru-Stop, Fig. 1, and the drum brake is called the Duo-Grip, Fig. 2.

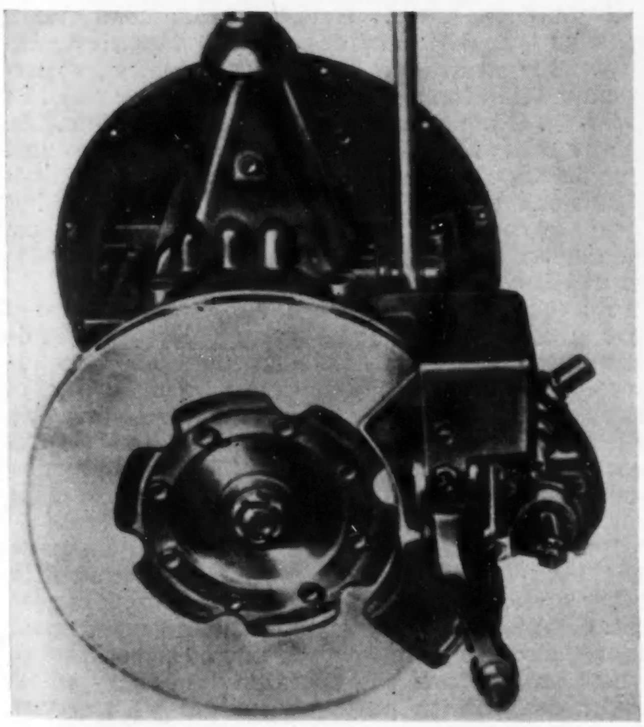
Because of its relatively larger diameter of rotating part, the disc brake introduces a more difficult dynamic balance problem under severe operating conditions. The Duo-Grip brake's drum is conventional in size and light in weight, so that the balance problem is not complex.

Although both these designs are physically unalike, their operating characteristics are similar in that both have high capacity by using maximum cooling from surrounding air. And drum expansion does not affect them since the shoes actually pinch the brake drum between them.

All these designs are effective as holding brakes, but vary considerably in repeated stopping ability.

Auxiliary brake location, either in the drive line or at the rear wheel, also poses problems.

The drive line location has the advantage of brake power multiplication by the drive gear train ratio. This varies from 4 to 8, depending on the type of vehicle. In using this power ratio many auxiliary brakes are mounted directly on the transmission gear case, as in Fig. 3. This provides a



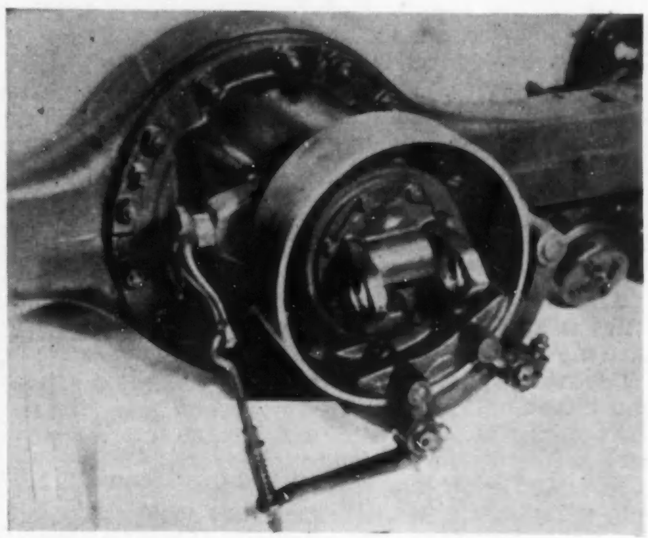


Fig. 2—This drum type of auxiliary brake is called Duo-Grip

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Fig. 1—Tru-Stop, a disc type auxiliary brake

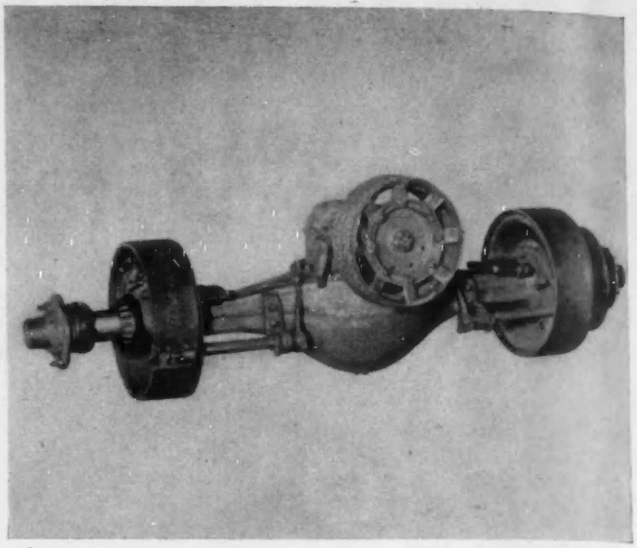

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Fig. 4—This auxiliary brake, mounted on the pinion housing of the rear axle, imposes high stresses on some parts of the drive system, but not on the propeller shaft and universal joints, as does the arrangement in Fig. 3

←

Fig. 3—Mounting an auxiliary brake on the transmission adds both little weight and cost. But it applies torque loads to the transmission gear case and transmission main shaft bearings

short coupling for manual control and generally makes for about as inexpensive and light weight a design as possible.

But this mounting has the disadvantage of introducing brake torque loads into the transmission gear case and transmission main shaft bearings. Such a condition is particularly serious with automatic transmissions and torque converters since the clearances are more critical than those on conventional gear boxes.

The transmission mounting also is farthest from the wheels so that brake torque is transmitted through the propeller shaft, gear set, axle shafts, and wheel drive. Failure of any one of these parts renders the brake ineffective. Introducing auxiliary gear boxes in the drive line further detracts from the transmission-mounted auxiliary brake due to disengagement possibility. Another point of censure is the greater fire hazard compared to other locations. Grease leakage from the transmission shaft is a contributing factor.

Evaluation of transmission mounting characteristics also apply to an auxiliary gear box or transfer case mounting of the auxiliary brake.

Desire to eliminate the transmission main shaft and gear case loading, due to the brake, has made the independent mounting on a frame cross member an acceptable alternate. This type mounting is suitable for trucks and buses, particularly where long wheelbase is available. But it still interposes the propeller shaft, gear train, axle shafts, and wheel drive between the brake and wheels.

A third type mounting widely used on rear-engine buses is the auxiliary brake installation on the pinion shaft housing. See Fig. 4. It eliminates the propeller shaft and universal joints from the group of parts subjected to high stresses during brake application. Propeller shaft failure is especially damaging on buses or other vehicles having several brake or control lines running parallel to the shaft. It is not unusual for a broken shaft to tear out these important lines.

Regulatory statutes in some states require guards or brackets around the propeller shaft to meet this emergency. These add cost and weight.

The rear axle drive pinion cage mounting eliminates stressing of the propeller shaft, and offers an advantage as such. But the brake load still is carried through remaining drive system elements.

With regard to stressing drive system parts, particular attention should be focused on the rear axle gearing. Ring gear and pinion are most vulnerable. Many believe that the same gear load values used for driving also can be used for braking. This is fallacious because it overlooks fundamentals of gear teeth design. Tooth loading and pressure angles are much more favorable on the drive side than the coast side of gear teeth.

All these considerations have induced continued use of the wheel brake as an auxiliary brake. Advantages of the wheel brake location are its proximity to wheel and tire. It completely eliminates braking load on the drive system and makes available the large capacity of the wheel brakes. Brakes run cooler, with fire hazard at a minimum. Common auxiliary and service brake shoes and drums make for a low cost system, with little added weight.

But it has the disadvantages of common parts and need for additional power, as compared with other brakes, due to inability to use the drive gear ratio. Operating mechanism, such as rods and cables, also are more involved due to remoteness of wheel brakes from the operator. Auxiliary wheel brake design with parts uncommon with service

brakes have been limited by two factors—space restrictions and the need for making service brakes as large as possible.

Auxiliary brakes mounted in locations other than the rear wheels complicate brake drum material and design problems. Heat checking tendencies and burst strength of cast-iron drums have given unsatisfactory operation, so that alloy cast irons or special cast steels are used advantageously. Stamped steel drums combined with suitable brake linings have secured high strength advantage from this metal in certain designs.

Recent auxiliary brake types have essential elements of good design. Perhaps wheel brakes can be used to better advantage as auxiliary brakes. Interconnecting the auxiliary and service brake systems to realize their design objectives—stopping the vehicle and holding it—points to best results.

A simple "parking brake" is inadequate for this purpose; but an auxiliary brake with good capacity, coordinated with the service brake system, will reap the most benefits and the greatest safety. The two brake systems should be maintained as independent and separate arrangements, as required by law.

A Correction

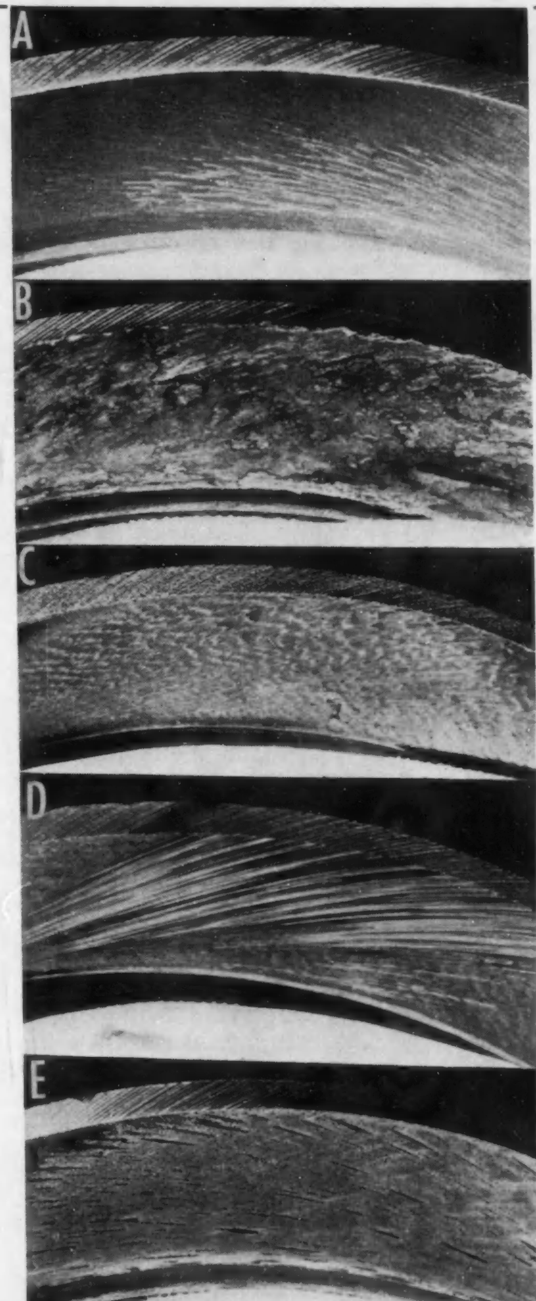
High Torque Test Of Gear Lubes

In Fig. 3 of the article "Gear Lube Testing Procedures Compared," by W. B. Bassett, The Lubrizol Corp., on p. 69 of the SAE Journal, June, 1949, the photos for "D" and "E" were transposed, making the caption incorrect. The corrected illustration is shown at right and the caption for it reads properly as follows:

■

This series of photos illustrates the results of a high-torque test. "A" shows the drive side of a new pinion tooth with its original surface finish. After running for 3 min on straight mineral oil after load application, it took on the appearance in "B." Note the places where welding took place.

Another type of failure, shown in "C," is described as a rippled surface; that in "D" is considered ridged. Both of these are said to stem from plastic flow of metal under the load and speed used in this particular test. The tooth in "E" is from a test on a 2–105B product and shows normal wear with no surface distress. By comparing this with the new gear in "A," you will recognize the dark irregular lines along the surface as tool marks.



"SAFETY FIRST" Air Transport

SAFETY is always the foremost consideration of those who manufacture and those who operate our transport aircraft.

First, untold safety is engineered into these planes by the maker, despite the high cost of such features, in terms of both weight and dollars. Then, other safety equipment is added by the airline when it receives the planes, before they are put to use in scheduled operations. Most of us, even those in the industry, just take these features so much for granted that we don't stop to realize the extent of their benefits.

Finally, there is the organized effort being made to study each accident that does occur, with the objective of preventing the recurrence of the trouble that caused it.

Safety Provisions

Safety features incorporated in air transports include the following:

1. More than one engine per aircraft, despite the fact that, purely from a performance standpoint, it would be more economical to operate our present 4-engine aircraft, for instance, with one engine having a capacity equivalent to the four smaller ones. Thus, the government regulation that requires at least two engines (so that one may continue to carry the loaded aircraft if the other one fails) is purely a safety measure. It is costing the airlines around $7500 and 750 lb of weight per airplane to provide this arrangement.

2. Elaborate provisions for extinguishing fires in the engine and baggage compartments. We hope that it will never be necessary to use them but, again purely for safety reasons, fire-fighting equipment weighing about 545 lb per airplane is added. If the airlines were not safety-minded, they would carry this additional load in revenue cargo.

3. Duplicates of practically every instrument on the instrument board of the cockpit. Basically, only one set is required to fly the airplane, the duplicate is carried for only one reason—safety—to take over if the first instrument fails. Instruments cost plenty of money, and they take up space. Such additional equipment costs an estimated $3000 per airplane and weighs about 100 lb.

4. Radio provisions, which are installed mainly for safety reasons. If we did not have the problems of weather and navigation to consider, we could eliminate radios, and the duplicate sets of radios, which are installed as standbys. Such radios represent a considerable weight and are very costly. In fact, the radio installation costs the airline more than $6000 per plane and weighs about 800 lb.

5. Propeller feathering equipment. This is a device for stopping the propeller blades in flight to avoid destruction of an aircraft engine that fails. Such equipment weighs 57 lb.

6. Fuel dump valves. These allow the pilot to drop his fuel load if he has to make a forced landing, in order to reduce the fire hazard. These valves add 146 lb to the weight of the airplane.

7. Two electrical batteries. One is all that is required, the other being a standby, for use if the main one stops. One battery removed would save 82 lb.

8. Automatic pilot. This is not a necessary flight instrument, but one that is installed to reduce pilot fatigue, thereby increasing pilot effectiveness and alertness, and thus add greater safety. The extra weight is 211 lb.

9. Hand fire extinguishers. These are distributed around the cockpit and cabin. They weigh 81 lb.

10. Flares. These are used in emergency night landings. They add another 44 lb to the load.

These are but a few of the many safety additions. They weigh a total of over 2816 lb.

For easy figuring, let us cut this to 2000 lb.

Each pound of potential payload is worth $40 per airplane per year. Then, 2000 × $40 = $80,000, which

* Paper, "Some Aspects of Air Transport Operation," was presented at a meeting of the Midcontinent Section of the SAE, Tulsa, Okla., March 18, 1949. (This paper is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

Is Motto Of Industry

BASED ON PAPER* BY

Otto E. Kirchner

Director of Engineering
Tulsa Maintenance Depot
AMERICAN AIRLINES, INC.

is the yearly loss in payload per airplane that the airline experiences because of the weight of the safety features. For a fleet of 50 airplanes, the cost would be $4,000,000 in lost payload per year.

No account has been taken here of the reduction in maintenance and engineering costs that would result from the deletion of such extra items.

What about Parachutes?

But what about parachutes? If airlines are so concerned with safety, why aren't parachutes carried? The fact is that they are not used in present-day aircraft because careful analysis, based on our investigations of past airline accidents, shows that there is but small possibility of parachutes being useful.

One cannot say, however, that they will never be added to our aircraft, for as soon as parachute designs and aircraft development prove that devices of this type will add a genuine safety factor, then they will be considered for air transport installations.

Investigation of Accidents

It is unfortunate that the airlines have accidents, but when they do, each accident is studied exhaustively to determine its causes and to develop means of preventing its recurrence. Each accident, for this reason, has increased our knowledge of how to make air transportation safer.

Our present knowledge of safety devices has the potentiality of overcoming such anticipated hazards that we think might still exist. It might surprise some people to learn the amount of energy that is put forth daily by all airlines to improve their equipment for safety and, incidentally, to reduce the cost of operation. It might also be news to them to learn that only seconds sometimes elapse between the time one operator experiences some condition of flight or equipment that conceivably might affect safety and the time that all operators are alerted and aware of the possible hazard.

If it is in the equipment, then each airline immediately makes a check on all similar airplanes to determine whether the same condition may be present. This coordination is all done through a central agency known as the Air Transport Association, whose one objective is improved and safer air transportation. If it is a flight condition, all pilots and ground dispatchers are alerted to avoid possible hazard.

In connection with safety, it is interesting to note that in flying, unlike all forms of ground transportation, an increase in speed does not mean an increase in the hazard. We know from experience in driving our own cars that surveillance, carefulness, watchfulness of the roadway, and competence of drivers become more and more important as ground speeds go up. Because of this factor, we cannot hope to see any appreciable increase in speed for ground transportation for some time to come.

With reference to aircraft, on the other hand, there is no instance of speed of flight having any bearing whatsoever on any accident ever investigated.

Dividends

Dividends that the airlines are obtaining as a result of all these safety efforts are taking the form of increased acceptance of air travel.

Because of the airplane, many a person is satisfying a wish that he could not otherwise have satisfied. For instance, it is not uncommon to find persons in their eighties going by air to visit their children in distant parts of the country. They would not have made the trip at all, it seems, if it had involved lengthy ground transportation because of the inconveniences, the nights on the road, and the physical discomforts of such a long trip.

It is surprising to note that 40% of the passengers are women with children, many of whom have plenty of time to spare, but who wish to avoid the nerve-racking tension that results when children are taken on long trips by means of ground transportation.

It is observations such as these that make it clear that air transportation is here to stay.

New CUT–WIRE Big Boon

By H. H. Miller

Research Metallurgist,
Buick Motor Division,
GENERAL MOTORS CORP.

EXPERIENCE with cut-wire shot for peening automotive chassis springs reveals these advantages for the material:

1. Cut-wire shot is an ideal peening material from the standpoint of uniformity of physical properties, size maintenance, and life.
2. It is available in many sizes and can be purchased in production quantities from different sources.
3. It improves the peening operation both in quality and cost.
4. Better methods for manufacture of the cut-wire shot should result in better quality and lower cost.
5. The introduction of cut-wire shot is certainly one of the greatest improvements that has been made in shot peening. It decreases considerably many of the objections to shot peening. It allows greater control of the process. It should result in a new attitude toward peening and probably will make possible many new applications.
6. It is possible that with improved equipment additional changes in the chemical analysis and physical properties of cut-wire shot will allow a further improvement in shot peening costs.

Cut-Wire Shot Described

Cut-wire shot is made from MB hard-drawn mechanical spring wire. This wire is drawn to controlled physical properties and is the stage in wire-making just prior to oil tempering or annealing. The chemical analysis of this material will vary with the wire size. This chemical analysis is as follows:

Carbon	0.45/0.70
Manganese	0.60/1.20
Phosphorus	0.045 max.
Sulfur	0.050 max.
Silicon	0.10/0.30

The shot is made by cutting the wire into lengths which are equal to the diameter. Up until the present time most of the experience has been shot made from 20 gage (0.0348 in. $\pm$ 0.001) wire, although larger and smaller sizes are available. The experience has also been confined to the hard drawn type of wire, although the wire is available in the annealed or oil tempered condition.

The wire is purchased on tensile strength and we specify 261,000 to 301,000 psi. Although the wire cannot be purchased to a hardness specification, our experience has been that the hardness is uniformly between 45 and 50 Rockwell C.

As supplied, the shot has sharp edges which make the new material quite abrasive. With new applications it is necessary to use a break-in period to remove the sharp edges. The cut-wire in our first application was blasted against scrap in the peening machine for 4 hr before starting production. In additional applications we have diluted the new shot with used cut-wire shot from the first applications. The necessary day-to-day additions of new shot are very small and cause no particular difficulty. For jobs where there might be more severe surface requirements, such as peening aircraft parts, it might be desirable to buy conditioned shot.

In use, the cut-wire shot deforms into a spherical shape of uniform size. Its value for peening lies in the fact that it can be purchased in a uniform size with controlled physical properties, it maintains size during use, and has remarkable life.

For comparison, Fig. 1 shows new S330 chilled cast-iron shot and new cut-wire shot.

Test Description

The coil suspension spring peening application was selected for the initial tests on cut-wire shot because this was Buick's major shot peening application and very complete shot usage and maintenance costs were available over several years' experience with chilled cast-iron shot. Table 1 gives the operating conditions for both chilled cast iron and the cut-wire test.

As shown in Table 1, changes were made in the operating conditions to accommodate the cut-wire shot. The wheel speed was reduced from 2000 rpm to 1775 rpm because of the increase in the average size of the peening particles. With the lower wheel

SHOT

to Peening

Peening automotive chassis springs with cut-wire shot has improved quality and lowered costs, says Miller in this article, based on a paper he presented at a recent meeting of the SAE Shot Peening Division, of the SAE Iron & Steel Technical Committee.

He notes that peening has been a costly and comparatively uncontrollable operation because of excessive peening shot usage, short shot life, formation of grit which prevents control and makes for high maintenance costs, and lack of size uniformity and quality of shot as furnished.

Results of tests with shot made from hard drawn wire, detailed here by Miller, indicate that the new material overcomes these drawbacks.

speed and larger size of the peening particles, it was found necessary to force-feed the shot into the wheel by means of an air jet. The air jet idea was worked out by Chevrolet Gear and Axle; by its use the pounds of shot per minute delivered to the wheel were increased from 145 to 300.

Before the test was begun, the machine was reconditioned and the chilled cast-iron shot was removed from the system. It has never been practical to remove every last trace of peening shot from a peening machine for this type of test and there was a slight carry-over of chilled cast-iron shot in the cut-wire shot test.

To maintain records, a form was furnished to the shop to record information on operating conditions and control. These forms were completed daily, and were in addition to records kept by the production department.

At the start of the test, 1500 lb of cut-wire shot were added to the equipment described in Table 1. Scrap was placed in under the wheels and the machine was operated for 4 hr without conveyor travel to round the sharp edges of the shot. After this break-in period, production was begun. Because it was necessary to add additional shot to fill the cavities of the machine, it was decided to mark the beginning of the test when the quantity of shot added to the machine represented only usage. This condition was reached in seven days. During this seven-day period, the shot was rounding although the springs showed a good surface from the beginning.

Arc heights, Faxfilm impressions of the peened surfaces, and shot samples were taken periodically in addition to the normal control of the operation. This test was continued for 65 two-shift days, when the machine was shut down for overhauling.

Shot Behavior in Breakdown

As a part of the investigation of cut-wire shot, it was considered desirable to determine what pattern was followed in the breakdown during use. Extensive sampling during the first part of the test furnished samples that show the method of rounding. Samples taken periodically during the test show the tendency to maintain a uniform size. The photographs in Fig. 2 illustrate the progressive cold working of cut-wire shot.

The new material, as shown in Fig. 2A, is added to the peening machine. In this stage the particles are quite uniform in size. The wire gage is within a $\pm$ 0.001 in. tolerance. Judging from experience to date, it appears that the length of the particles can be maintained well within $\pm$ 0.0035 in., the figure used in the Buick specification for cut-wire shot.

As received, the material has sharp edges which will abrade the work and machine in an undesirable manner if the new shot is added in large quantities.

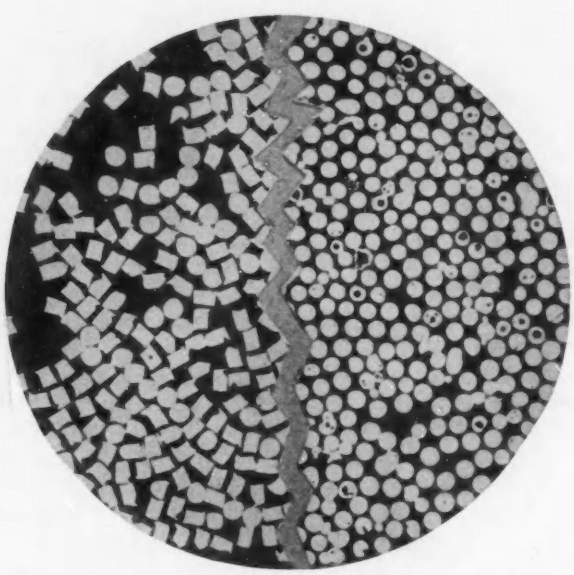


Fig. 1—A sample of new cut-wire shot, 0.035 in. (left) compared with new chilled cast-iron shot, size S330. Both are magnified three times

Table 1—Operating Conditions for Shot Peening Tests

Material

Coil Spring Material	SAE 9260
Material Size	Varying with application (0.568–0.725-in. diameter)
Sequence of Processing	Heat in walking hearth furnace—Coil—Fixture Quench—Temper—Shot Peen—Load Test (Breakdown solid, compress to specified length and measure load.)
Average Hardness of Coil Spring	51 Rockwell C
Weights of Coil Springs	9 to 13.5 lb (approximate)

Equipment

Type of Peening Equipment	American Wheelabrator Equipment Corporation
Production	450 springs/hr/machine
Number of Wheels	2
Diameter of Wheels	19½ in.
Width of Wheels	1⅝ in.
Distance of Wheels from Nearest Surface of work	15 in.
Work Rotation	54 rpm
Conveyor Speed	17.14 fpm

Peening Shot

	Chilled Cast Iron	Cut Wire
Size of Shot Added	230 and 330 (approximately 50% of each)	0.035-in. diameter × 0.035-in. long
Shot Flow	145 lb per min	300 lb per min
Wheel Speed	2000 rpm	1775 rpm
Shot Consumption/Spring	0.134 lb	0.013 lb
Arc Height	0.0155A2 average	0.0166A2 average

To take care of this difficulty, the shot must be conditioned on new applications before production is started. Fig. 2B shows what happens to the shot during a 4-hr conditioning period. The remaining photographs in Fig. 2 show the progressive rounding of the shot by hours up to 14½ hr and show the condition at 7, 28, and 68 days.

These photographs were taken of unscreened samples which were secured from material being delivered to the wheel of the peening machine. An effort was made to clean the machine of all chilled cast iron before the test was begun, but note the presence of chilled cast iron in Figs. 2B, 2C, and 2D. Chilled cast-iron grit in small quantities was found in shot samples very late in the test.

To determine what happened to the cut-wire shot in breakdown, a sample was screened and the material retained on each screen was mounted, polished to expose the cross-section, and photographed. These photographs are shown in Fig. 3. At first it appeared that the cut-wire shot developed hollow centers as it wore to smaller sizes and the shot resembles hollow chilled cast-iron shot. A closer examination, however, explains this phenomenon.

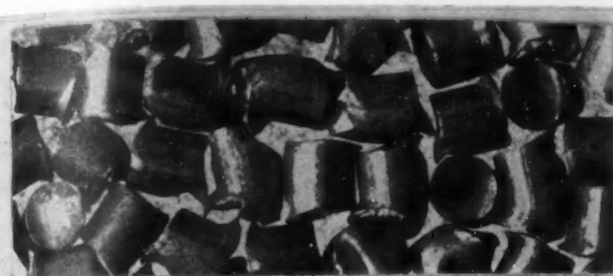
A micro-examination of the shot in Fig. 3 shows that the shot retained on the 0.033-in. screen, Fig. 3A, has been cold worked to a depth of 0.002 to 0.003 in. below the surface. This material does not show the presence of "hollow" shot. The shot taken off the 0.0197-in. screen has received considerably more cold work. Sections cut through the center of this shot show cold work extending from the surface to the center.

Other shot from this group is ragged and torn at the edges, with voids in the center which give the appearance that the centers pulled out. It is apparent upon closer examination that these hollows are caused by the edges of the broken particles peening over in a secondary rounding. On looking at a plane through this peened-over section in which the sample was polished, it appears that shot is actually hollow at the center, but the hollow is very shallow.

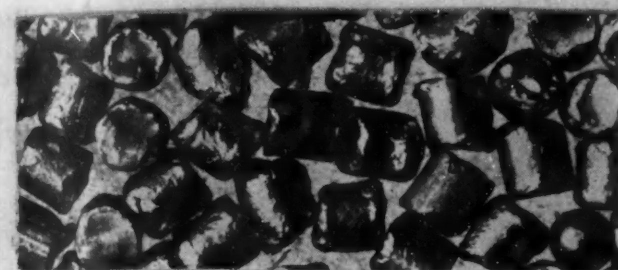
Fig. 4, which is a photograph of the shot used for the cross-section in Fig. 3C, clarifies this explanation further. The peened-over edges can be seen very clearly in many of the particles.

The shot taken from the pan, Fig. 3G, is very heavily cold worked and is either very irregular in shape or appears like splinters that have chipped off as the shot was wearing down. This sample contained a considerable amount of chilled cast-iron grit.

Figs. 5A and 5B show etched cross-sections of cut-wire shot in two stages of cold work. The rounding



A — New cut wire as added to machine



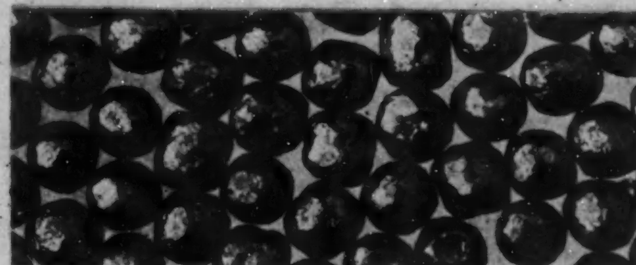
B — After the 4-hr conditioning period, during which the shot was thrown against scrap to remove sharp edges

C — After 5 ½ hr, which includes 1 ½ hr on peening springs

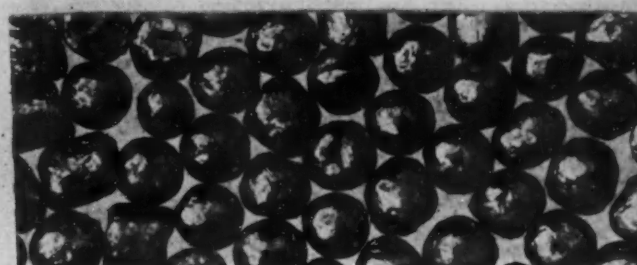
D — After 6 ½ hr, including 2 ½ hr on springs

E — After 11 ½ hr, including 7 ½ hr on springs

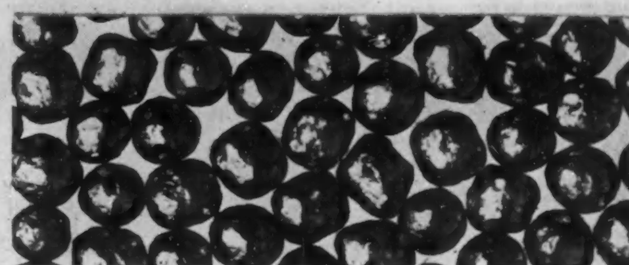
F — After 14 ½ hr, including 10 ½ hr on springs



G — After seven days



H — After 28 days



I — After 68 days

Fig. 2—These photographs—ten-time magnifications—show progressive cold working of cut-wire shot. Before putting fresh, unused cut-wire shot to work, it must be conditioned for about 4 hr to remove its sharp edges, if added in large quantities. Continued use after that, as these photographs show, progressively rounds the shot

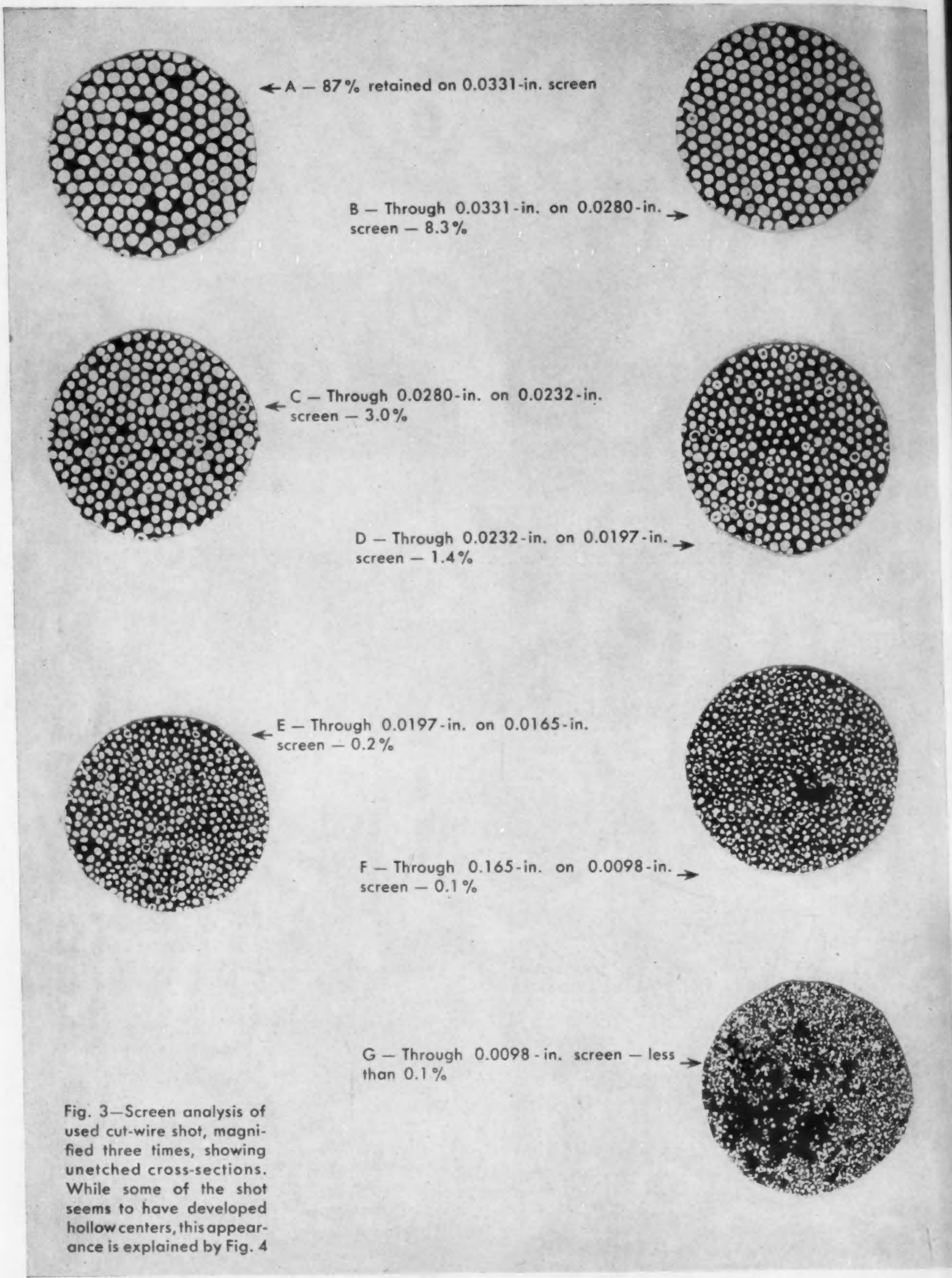


Fig. 3—Screen analysis of used cut-wire shot, magnified three times, showing unetched cross-sections. While some of the shot seems to have developed hollow centers, this appearance is explained by Fig. 4

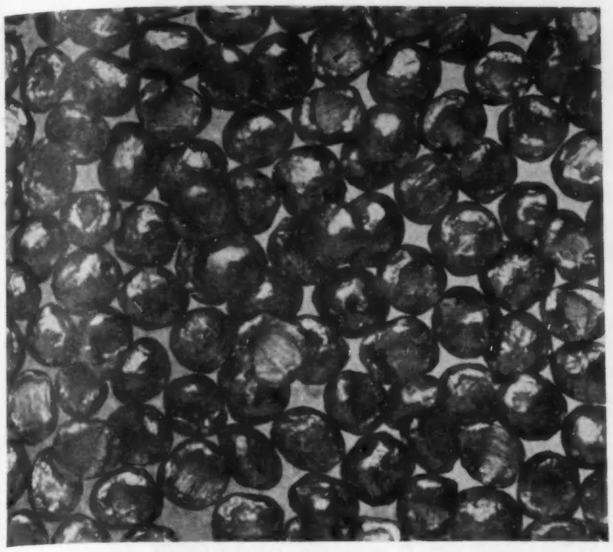


Fig. 4—This is a photograph of shot from Fig. 3C, magnified 10 times. The edges of broken particles, peened over in a secondary rounding, shown here, account for the "hollow" shot in Fig. 3

of the shot is apparently a combination of cold deformation and breaking off of the cold-worked edges. Fractures in the shot start at the surface and the shot breaks in the direction of the grain flow lines. We have not been able to find internal ruptures in the cold-worked shot.

Although the surface hardness of the shot may be increased in hardness considerably by use, we have found no method of determining what the increase might be. However, hardnesses taken on the cross-section of the used shot have shown no more than a one point Rockwell C increase. The method and control of cutting wire shot will undoubtedly have an influence upon the performance. An effort will have to be made by the manufacturer to cut the shot with a minimum amount of deformation and without the formation of shear cracks.

It is our practice to life-test a sample from each shipment and we believe that such a test should become a part of a specification for cut-wire shot.

Test Results

Before shot peening was used, the Buick coil chassis springs were made from four-pass ground bars. This amount of grinding was believed necessary to eliminate surface defects. The fatigue life averaged less than 100,000 cycles. With the introduction of shot peening it was found possible to use one-pass ground bars. In addition to the reduction in the amount of grinding necessary, it was also possible to reduce the amount of material in the spring. The fatigue life increased to an average life of slightly less than 500,000 cycles using chilled cast-iron shot.

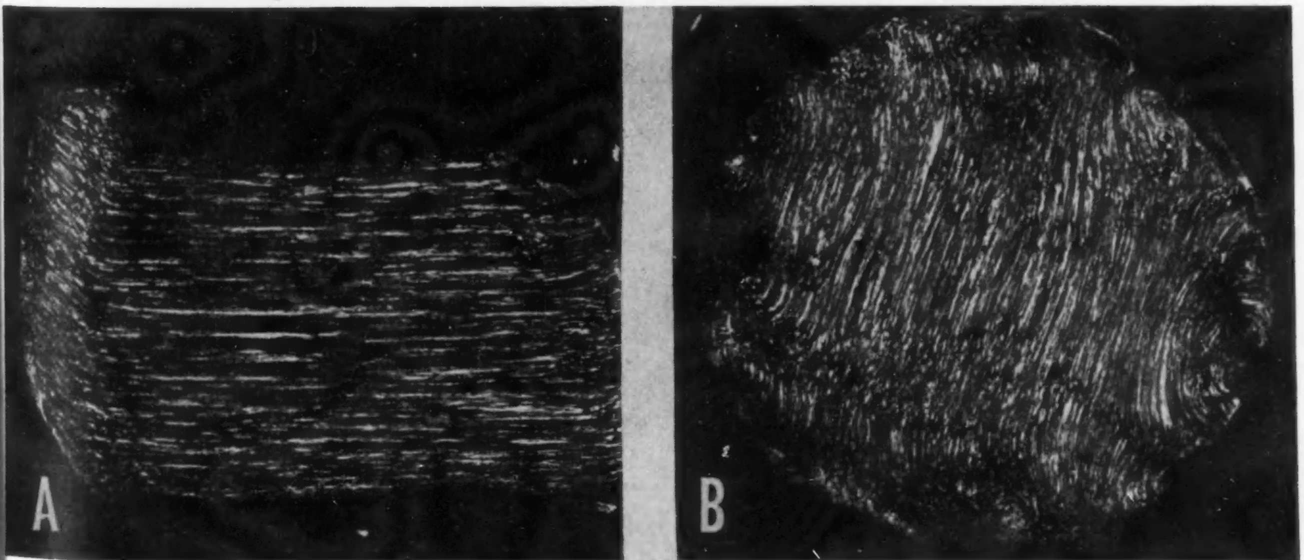
With cut-wire shot and the conditions as outlined, the fatigue life has been increased to a life in excess of 1,000,000 cycles. Our tests are discontinued at 1,000,000 cycles when no breakage occurs. With this improvement in life it may be possible to make further savings in processing costs.

The extent to which a surface is covered with shot impressions or coverage has a bearing on the quality of the peening job. But the extent to which a peened surface is covered with impressions does not necessarily indicate the amount of compressive stress introduced.

In the past, even with the use of separators, a considerable variation in the size of the peening material was encountered. This variation ranged from control of the size as purchased, by means of separators, to no control other than by dust collector. In the latter case visible coverage is better, but the peening job is probably not as good.

The ideal situation is probably to have 100% coverage with a shot of uniform size. However, if satisfactory minimum fatigue life can be obtained with a lesser coverage, it is a waste of money to specify the ideal condition. We do not peen for appearance, but to secure a satisfactory service life for

Fig. 5—These cross-sections of cut-wire shot, etched in 5% Nital, show cold working of the surface. They are magnified 100 times. "A" shows slightly used cut-wire shot, and that in "B" is used cut-wire shot



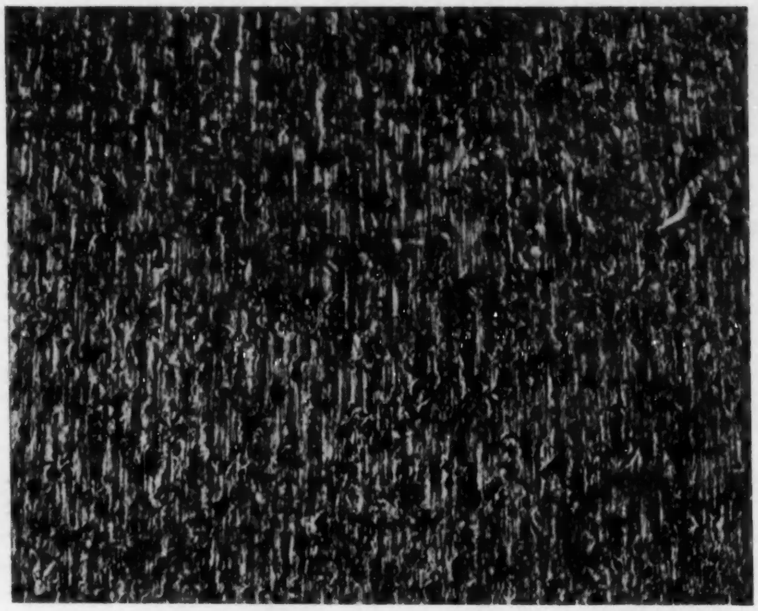


Fig. 6—Surface of a Buick coil spring peened with cut-wire shot, magnified 10 times

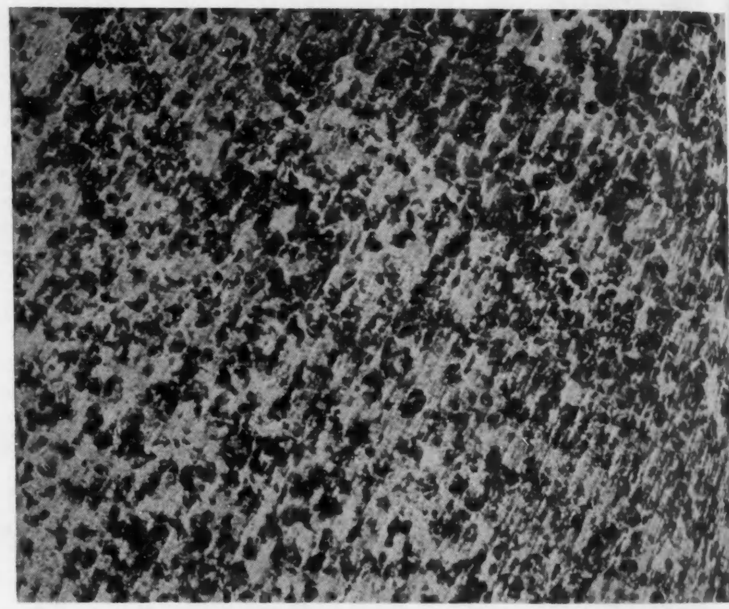


Fig. 7—Faxfilm impression of the same surface shown in Fig. 6 (a coil spring peened with cut-wire shot)

the part. Each application must be proved on its own set of operating conditions.

Because of the greater average size of the cut-wire shot it was necessary to increase the shot flow to secure coverage. Visible coverage is probably reduced from that obtained with chilled cast-iron shot because of the large percentage of small particles present. No attempt other than test has been made by Buick to use the shot separators either with chilled cast-iron shot or cut-wire shot. The percentage of fines using cut-wire shot is very small and we believe the dust collector is an adequate means for maintaining clean shot.

Fig. 6 shows a photograph of the surface of a Buick coil spring peened with cut-wire shot. Fig. 7 shows a Faxfilm impression of the same area.

Shot Usage

In this test, cut wire shot usage was approximately one-tenth of the usage experience with chilled cast-iron shot. A large percentage of shot is lost through waste during the peening operation. It is difficult to determine the percentage lost by wastage; but it can be seen from a comparison of the results found with shot testing machines with production results that shot losses must be a great percentage of the usage. Cut-wire shot will show an advantage of 250 to 300 to 1 over chilled cast iron shot in the test machine, but production experience shows an advantage of only about 10 to 1.

It is true that in a production machine the shot has many impacts per cycle but this cannot account for more than a small part of the difference in results. In peening coil chassis springs the carry-out losses are at a minimum; but despite efforts to stop leakage to the outside, there was considerable shot on top of the machine and on the floor in the immediate vicinity of the machine. This shot was returned to the machine as well as could be done practically. It is certain that much work must be

Table 2—Cost Comparison of Chilled Cast-Iron and Cut-Wire Shot for Peening Chassis Coil Springs

	Chilled Cast-Iron Shot	Cut-Wire Shot
Number of Springs Shot Peened*	1,000,000	1,000,000
Bags of Shot Used—100 lb per Bag	1342	131.97
Shot Cost per Ton†	$110.00	$400.00
Number of Springs per Pound of Shot	7.45	75.77
Shot Cost per Spring	$0.0073	$0.00255
Average Arc Height—Almen A2	0.0155 in.	0.0166 in.
Maintenance Labor	$687.74	$498.93
Maintenance Material	$2994.33	$1451.37
Maintenance Cost per Spring	$0.00366	$0.00195
Total Cost of Shot plus Maintenance per Spring	$0.01096	$0.00450

* The actual number of springs compared was 1,000,528 for the chilled cast-iron shot and 447,076 for the cut-wire shot. The comparison was made on the basis of 1,000,000 springs for ease in reading the table.

† This is not the current market price for either material, but represents costs at the time that the test was carried out.

done with peening equipment so that shot can be used to obtain the benefit of the longer life that is being developed.

A sample of the shot was taken daily from the machine for screen analysis. This sample was taken from shot being delivered to the wheel and was taken at the same time every day. The period when the sample was taken was 8 hr of operation from the time that the last shot was added to the machine. No samples were taken for screen analysis during the first seven days of the test.

The size was found to remain uniform throughout the 80-day period charted and the size is stabilized from the seventh day on. The amount of shot added to the machine each day after the tenth was 100 lb. Screen analysis showed that an average of 80% of the material being delivered to the wheel is maintained on the 0.033-in. screen. About 15% is retained on the 0.028-in. screen, and more than 98% is retained on the 0.033, 0.028, and 0.023-in. screens.

There is a marked decrease in machine maintenance costs when cut-wire shot is used. This is explained by decreased hardness and absence of sharp grit in stabilized cut-wire peening material. The machine used for this test ran approximately twice as many springs before major overhaul was necessary than was experienced with chilled cast-iron shot. Blade life was at least 10 times as good. Other machine parts showed a decrease in replacements.

Cost Advantage

Table 2 shows a cost comparison of chilled cast iron and cut-wire shot for peening chassis coil springs. The comparison was made on the basis of 1,000,000 parts. The savings possible using cut-wire shot, depend upon the application. The cost advantages over other peening materials are explained by its long life which reduces usage, and handling costs. Because of the physical characteristics of cut wire shot, maintenance costs are reduced considerably. In addition to these savings in operation, an improved peening job is secured, and it may be possible to increase stress and thus make productive material savings.

Because of the greater initial cost of cut-wire, it is necessary to investigate each new application thoroughly and to reduce wastage as much as possible, both from carry-out and loss through the machine. Because of the greater tendency to maintain a uniform size, it will generally be possible to reduce wheel speeds and thus secure an additional advantage in the reduction of shot usage and maintenance costs.

New Cut-Wire Shot Uses

At present, Buick has three chassis coil spring peening machines operating with cut-wire shot. Another interesting application that is now in production is the cleaning and stressing of the inside surface of the Dynaflow brake bands with cut-wire shot. This operation was previously carried out with chilled cast-iron grit. The usage was decreased from 6000 lb of grit per week to 60 lb of cut-wire shot per week. Maintenance costs were reduced by an estimated 80%.

The Buick axle shaft is being cleaned with cut-wire shot. Although this is primarily a cleaning operation, some benefits are present from peening. This is in the testing stage.

Arrangements are being made to blast the Buick clutch spring with cut-wire shot. Because it is necessary to use two sizes of shot in this operation with the present operating condition, plans are being made to install a variable speed wheel so that only one size shot will be necessary. Other applications for peening and cleaning are being investigated to determine the advantages possible with the use of cut-wire shot.

Turboprop

A B-17 with its nose modified to carry a test engine is proving highly successful as a flight test vehicle.

Although designed especially to test the Wright Typhoon turboprop (T35-1), it is readily adaptable to a wide range of engine types, both reciprocating and non-reciprocating. In fact, it represents the closest possible approach to an all-purpose test airplane for engine development work. (See Fig. 1.)

Behind this final success lies an engineering program involving three separate problems:

1. Selection of the most suitable airplane.
2. Design of fuselage modifications to accommodate the added weight and stress requirements with the extra engine installed.
3. Design of the powerplant installation.

Airplane Selection

The B-17 was selected as the test vehicle on the basis that it offers the best compromise between flexibility and economy of flight testing, safety of experimental test operation, adaptability to a nose installation of a fifth engine, and ability to operate at moderately high altitudes for a sufficient period of time.

Fuselage Modifications

Fuselage modifications, which were made by the Boeing Airplane Co., consisted of moving the cockpit location rearward 4 ft, providing a fireproof powerplant bulkhead with the necessary mount attachment points just ahead of the pilot's and copilot's feet, reenforcing the fuselage structure back to a point in the tail where the ballast was to be located, adding a flight engineer's station in the bomb bay, and covering the bomb-bay doors with stainless-steel fireproof skin for about 10 ft.

The flexibility of this arrangement as an engine test bed is well illustrated by the fact that the airplane is capable of flying with no fifth engine, that is, merely with a nose fairing, or with any propeller type of powerplant weighing up to 4 tons.

Confirming early Boeing predictions, the B-17 with T35 turboprop installed showed no change in aerodynamic control characteristics over those of the conventional B-17; nor did the substantial overlap of turboprop and inboard propeller discs cause undue vibrational effects at low turboprop power when the inboard reciprocating engines were also running. Normally, turboprop testing was conducted with the two inboard engines feathered.

Powerplant Installation

The job of designing the powerplant installation can be broken down as follows:

1. Engine mounting system.
2. Powerplant exhaust system.
3. Turboprop air entry.
4. Lubrication system.
5. Fuel system.
6. Ventilation.
7. Starting system.
8. Pressure reading system.

Engine Mounting System—After considerable deliberation, the mounting system selected consisted of a built-up stainless-steel monocoque cylindrical section, extending from the airplane firewall attachment points to the turboprop center flange location, as shown in Fig. 2.

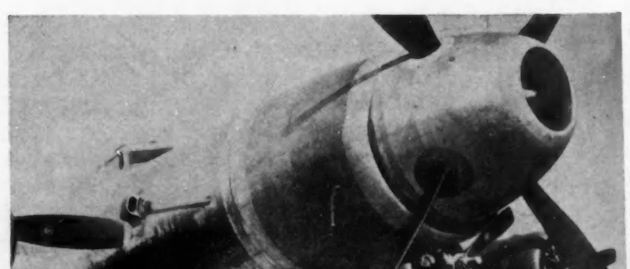


Fig. 1—Turboprop installation on B-17

Tested in Flying Lab

BASED ON PAPER* BY

R. R. Templeton and M. P. Cervino

Wright Aeronautical Corp.

This construction gave minimum installation weight and maximum safety from fire. It has proved satisfactory, aside from its limited maintenance accessibility, an inherent disadvantage of any monocoque structure.

Initial flight tests, conducted with the engine solidly mounted, showed no abnormal vibration but, unfortunately, they were not extensive enough to show conclusively whether or not the solid mounting would be feasible.

Most of the flight tests have been conducted with a dynamic mounting system, designed with particular regard for any undesirable vibrations developed by the propeller.

The turboprop, in common with the turbojet, normally develops no internal exciting forces that cannot be isolated; however, the dynamic links evolved became quite large (as shown in Fig. 3), in order to isolate propeller first-order vibration excitations of 400 cpm and above.

* Paper "Flight Testing the Wright Typhoon Turboprop" was presented at the SAE National Aeronautic Meeting, New York City, April 11, 1949.

Powerplant Exhaust System—Initial design calculations indicated that with the soft dynamic mounting, considerable motion would take place in the plane of the turboprop exhaust outlet, which consists of an annular passage divided into six equally spaced segments, as shown in Fig. 4. No flexible metal joint is readily adaptable for use at this point, due to the outlet shape.

In addition, the high ground idling rpm of the turboprop seemed to indicate the need for forced cooling to take care of the hot oil coming from the propeller reduction gears during this period.

Both these problems were solved by using an exhaust gas ejector, which aspirated oil cooling air and served as a floating junction between engine exhaust outlets and the collector proper. (See Fig. 5.) This permitted mounting the dual outlet exhaust manifold from the fixed mount structure in such a way as to have no contact with the engine except for two spring-loaded sheet-metal rings, which provided fireproof seals, as shown in Fig. 6.

Initial tests revealed, however, that this arrangement was not functioning properly, due to the build-up of high exhaust pressures at the top of the col-

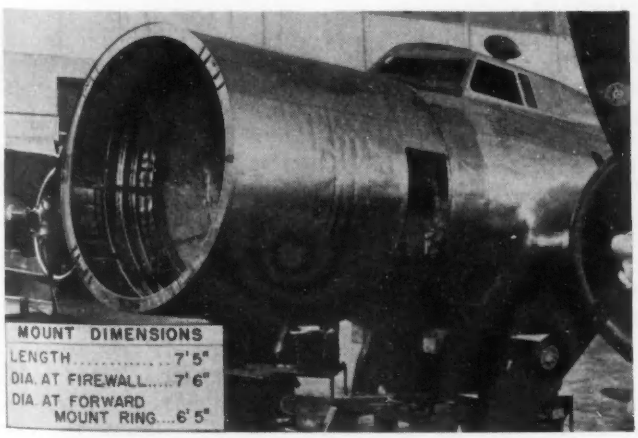


Fig. 2—Monocoque engine mount

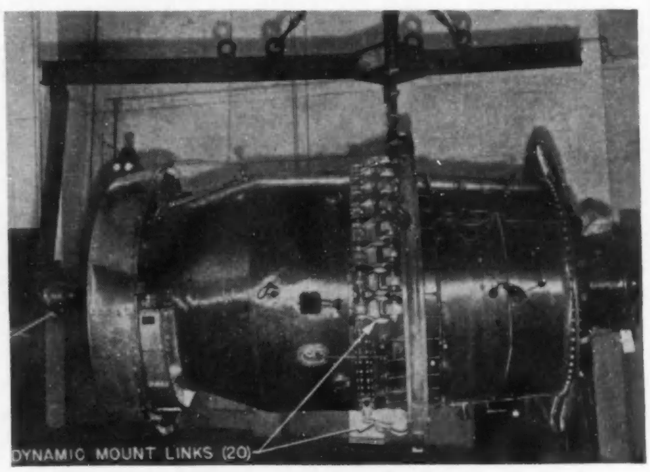


Fig. 3—Dynamic mount links

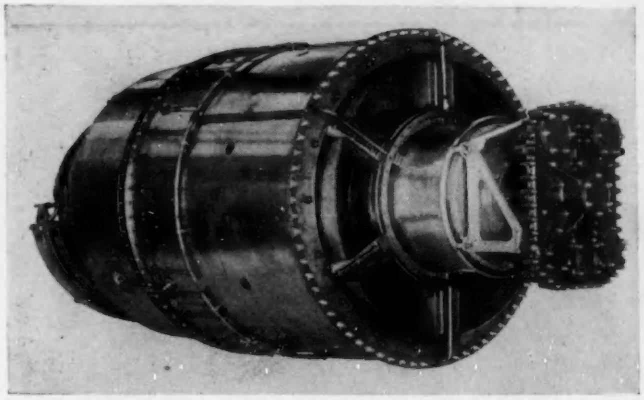


Fig. 4—Three-quarter rear view of Typhoon turboprop (T35-1)

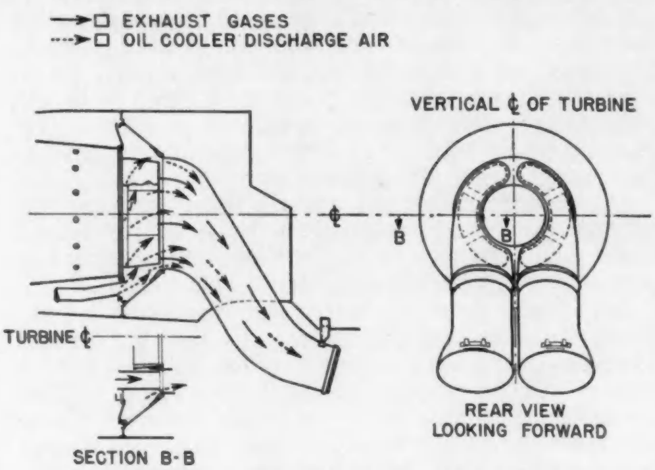


Fig. 5—Exhaust system air ejector (initial design)

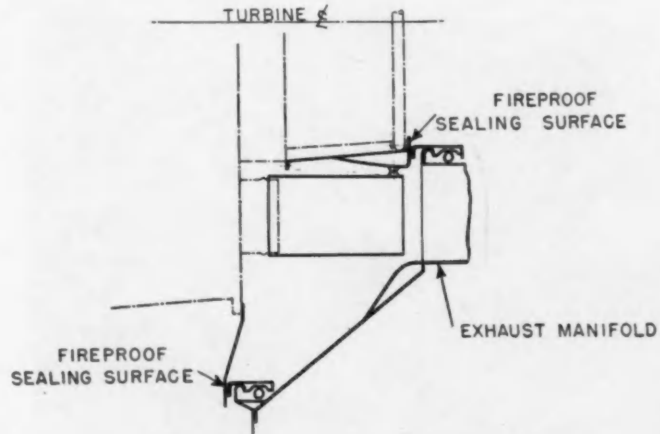


Fig. 6—Cross-section of flexible seals

lector (as illustrated at the left in Fig. 7), which caused recirculation of the gases at the ejector throat. It was believed to be due mainly to the sharp turn in the exhaust manifold. Incorporation of two turning vanes in each manifold leg created a more favorable duct aspect ratio in the region of the sharp turn and corrected the condition. The resulting change in gas flow pattern is indicated in Fig. 7. The turboprop exhaust outlets were also extended several inches farther into the ejector throat section, as shown by the heavy lines. Subsequent flight tests demonstrated complete equalization of static pressures in the manifold as well as restoration of the desired ejector pumping action.

Careful attention was given to fire prevention in the area of the exhaust system. Stainless steel completely shrouds the Zone 2 area (Fig. 8), as well as the peripheral surfaces, from the fuel nozzle location aft of Zone 2. Adequate CO_2 systems cover all zones susceptible to fire.

The exhaust outlets were installed at an angle of 15 deg from the adjacent fuselage line (Fig. 1). Fuselage skin temperatures in the area aft of these outlets did not reach 300 F. at any point, even during ground runup conditions.

Turboprop Air Entry—The basic premise was that any ram recovery less than 90% of free stream impact pressure would result in an unacceptable power loss under high-altitude, high-speed design conditions. Past experience indicated the NACA "E" cowl (Fig. 9) would give optimum performance for the T35 installation. A wind-tunnel investigation of a half-scale model of the cowling and engine air entry configuration was made under USAF sponsorship to check this prediction. The tests were run in the NYU 7 × 10-ft closed-throat, low-speed tunnel.

Results were extremely encouraging and correlated well with subsequent flight test data.

Early predictions by some that the size and possible complexities of the "E" cowl type of spinner would be a serious problem were not borne out by flight test experience. No vibrational, servicing, or other difficulties have developed.

Lubrication System—This system assumes a minor role in the installation, as it is an integral part of the T35 itself. Thus, the oil cooling problem was one merely of ensuring that the design cooling airflow was actually being provided, along with adequate manual airflow control to obtain test data over a range of airflow values.

The special annular type of oil cooler developed initially to permit reduced installation frontal area has been discarded in favor of the standard elliptical cooler.

Fuel System—The design is very simple, as it closely follows the conventional injection system used in reciprocating engines. Two separate tanks hold fuel for the fifth engine, each being fitted with an electric centrifugal boost pump. Fuel flows from these tanks through a selector valve to a mechanically driven boost pump, which is also an emergency unit, being mounted on the engine-driven remote accessory gearbox. A filter is located after this pump, just before an electrically controlled firewall shutoff valve. From here, the fuel goes to the engine.

Ventilation—Satisfactory ventilation of the turboprop outer shell was obtained by an inwardly opening flush cowl shutter (as shown in Fig. 8) at the top center and well forward, plus a cenventional cowl exit flap just ahead of the forward Zone 2 bulkhead and on the bottom center of the monocoque mount. Temperature control in the Zone 1 area was achieved through manually controlled flap actuator mechanisms on both flaps.

Two small airscoops, one on each side, in the plane of the horizontal centerline (Fig. 1) join together and return forward to an auxiliary cooling air blower integral with the engine and used to force circulation of cooling air through internal turbine parts.

Starting System—The starting arrangement (Fig. 10), while crude in appearance, is sound in principle and well suited for flight test work. It is composed of readily available nonaircraft parts, and so is cheap and quick to make.

A hydraulic pump develops 2500-psi pressure when driven by a 150-hp Packard engine. Fuel supply and return lines are routed to a similar hydraulic starting motor mounted in the airplane. An air-operated clutch is used between the turboprop and hydraulic motor to engage and disengage for starting. Quick disconnect fittings are provided on each external hydraulic line at the fuselage junction box.

With this rig, self-sustained engine running occurs within 30 sec or less.

In the air, windmilling starts are very smooth, clean, and rapid. There has been no evidence of flaming in the tailpipe during either starting or stopping of the engine.

Pressure Reading System—A simplified flight test pressure reading system was designed that reduced the weight and complexity of pressure reading instrumentation by more than half. Briefly, the system employs the principle of a remotely controlled, pneumatically actuated selector switch at the engine. Large quantities of tubing ordinarily routed over long distances can thus be replaced by a fraction of the original quantity of tubes and gages at the recording station in the cabin. The success of the system stems from its absolute reliability, economy, and instrumentation weight reduction.

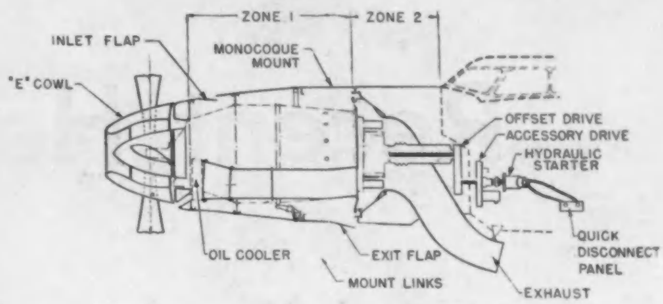


Fig. 8—Powerplant installation

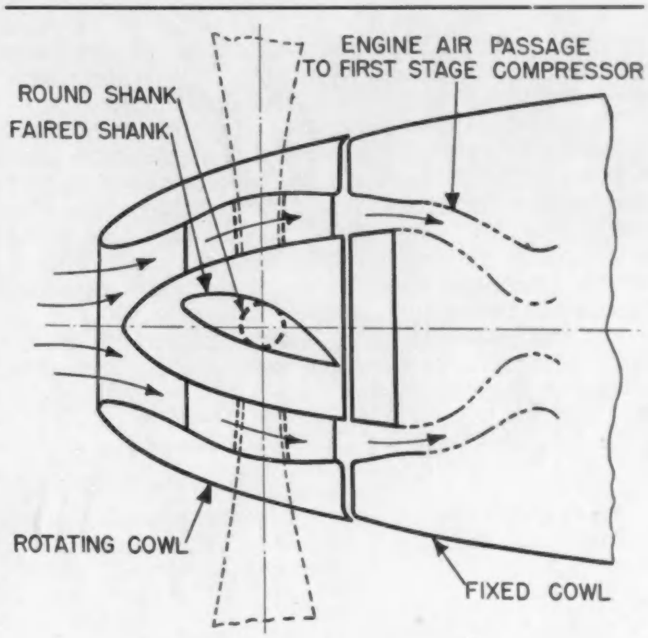


Fig. 9—Cross-section of NACA "E" cowl

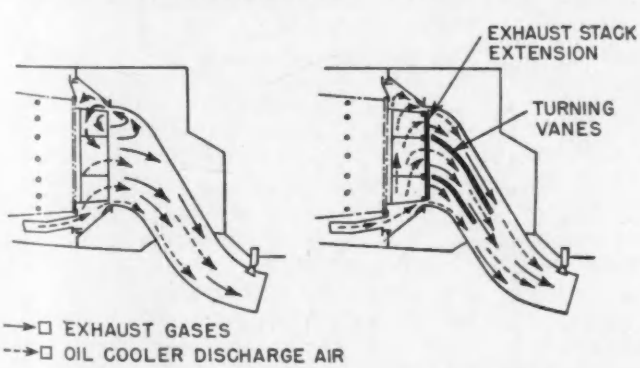


Fig. 7—Exhaust system—left: original; right: modified

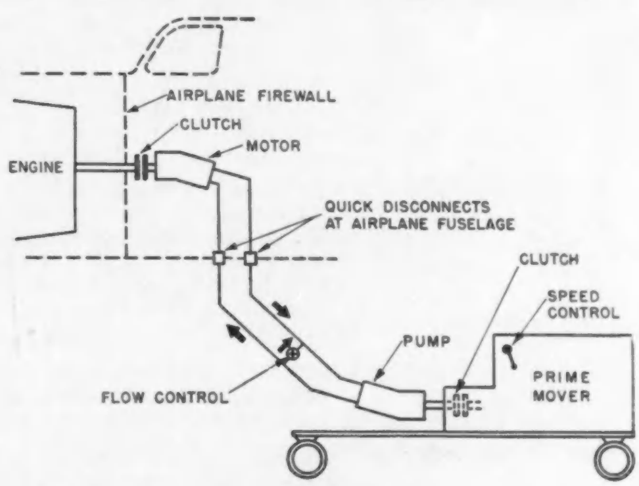


Fig. 10—Schematic diagram of starting arrangement

Recent Heat–Treatment

Heat-Treatment Atmospheres

By O. E. Cullen

SURFACE COMBUSTION CO.

DURING the past few years efforts have been made to improve protective atmospheres and to provide simplified means for controlling them.

These efforts were not confined merely to production of new heating furnaces and atmosphere generators. Equilibria data have been reviewed to find the correct equipment and best atmosphere for each specific use. This made possible what amounts to production line heat-treating, with a nicety of control comparable to the most exacting laboratory techniques.

For most practical purposes, a limited number of basic atmospheres need be considered. These are shown in Table 1.

Gray Iron Heat-Treatment

By C. F. Joseph

CENTRAL FOUNDRY DIVISION, GMC

HEAT-treating gray iron improves its machinability by eliminating the pearlite and forming a ferritic matrix.

Reason for poor machinability, prevalent today, stems mainly from nonuniformity of castings. Hardness and microstructure vary, due principally to lack of good raw materials and poor metallurgical control.

Matrix of a gray-iron casting is often sorbitic or sorbitic pearlitic in structure, with a Brinell hardness of about 217 to 228. Heat-treating most plain carbon gray irons at 1300 to 1400 F for 45 min, followed by slow cooling, will lower the Brinell hardness from 228 to the 140 range. If there are no

Table 1—Basic Heat-Treatment Atmospheres

Type of Atmosphere	Method of Preparation and Composition	Uses
Exothermic Base Gas	Prepared by combustion of air-gas mixtures and consists of 5% carbon, 10% carbon monoxide, 10% hydrogen, fractional % methane, and some water vapor and the balance nitrogen	Brazing low carbon steel; cycle annealing forgings; and any steel heat-treatment requiring non-scaling atmosphere where decarburization is no problem
Endothermic Base Gas	Catalytically-cracked mixture of hydrocarbon gas and air; product gas analysis shows about 20% carbon, 40% hydrogen, 40% nitrogen, fractional % methane, and traces of carbon dioxide and water vapor	Clean hardening medium and high carbon steels; reheating after carburizing; brazing high carbon steel and tool steels; carbon restoration and homogeneous carburizing (which is specially adapted to thin-section parts, such as stampings and tubing, requiring deep drawing); enriched with natural gas or other hydrocarbons, it provides an excellent carburizing atmosphere; also useful for suspended carburization (to stop furnace for 48 hr or longer)
Charcoal Base Gas	Prepared by passing air or moisture-free products of gas-air combustion through hot charcoal bed; analyzes about 34% carbon monoxide and 66% nitrogen or about 20% carbon monoxide and 80% nitrogen, depending on method of preparation; small amount of hydrogen present as impurity	Clean hardening and basic gas for carburizing and dry cyaniding
Nitrogen Base Gas	Prepared by removing carbon dioxide and water vapor from exothermic base gas; gas is substantially inert	Large-scale annealing processes
Ammonia Base Gas	Made from anhydrous ammonia, this is a highly-reducing mixture consisting of 10 to 75% hydrogen and 90 to 25% nitrogen, depending on method of preparation	Large-scale annealing processes

Developments*

chilled edges at this hardness, combined carbon is practically eliminated and machinability at its best.

Results from annealing gray-iron castings usually more than justify the cost.

Annealing to produce soft-gray-iron castings in some cases doubled the output of present machines, eliminating the need for additional capacity. On some jobs tool life has more than tripled. This together with need for fewer men to maintain tools add up to sizable savings in most instances.

Induction Heating

By A. W. Herbenar

OHIO CRANKSHAFT CO.

INDUCTION heating for forging and heat-treatment already is bringing savings to users despite its infancy.

These six savings are inherent in induction heating for forging; (1) accelerated heating minimizes scaling so that scaling loss and cleaning costs are reduced, closer tolerances are permitted, and die life increased; (2) start-up costs and costs during down time are saved; (3) it permits closer control of temperature and localized heat zone; (4) less floor space is required; (5) working conditions are improved; and (6) it is more readily adaptable to automatic operation.

There are many cases where induction heating is being used for forging. For example, one automotive parts maker is using a special remote induction heating station to upset automatically and to blank manually two ends of a bar simultaneously. The bars are hopper fed through two channel-type inductor coils which heat about 3 in. of each end to required forging temperature.

A transfer mechanism then transfers the bar to a double upsetter, provided the Rayo-Tube temperature control has not rejected the piece due to improper temperature. After upsetting both ends simultaneously, the piece drops into an underlying conveyor which delivers it to a press for a manual blanking operation.

Induction heating currently is being used most prominently for selective surface hardening where its biggest attraction lies in adaptation to automatic handling. Limited floor space requirements of induction hardening units makes it possible to install them directly into the production line, cutting transfer costs and increasing productivity.

A noteworthy example here is an axle shaft supplier who is progressively hardening them on an automatic induction heating machine. Hardness of 50 to 55 Rockwell C of the case plus favorable stress distribution impart exceptionally good torsional yield strength to the shaft. Residual compressive stresses in the hardened case increase torsional fatigue strength 200% over conventionally hardened and drawn axle shafts. Induction heating derives savings in this case by eliminating alloy requirements and by reducing straightening and machining costs.

Salt Bath Heat-Treatments

By L. B. Rosseau

AJAX ELECTRIC CO.

THE three characteristics peculiar to the salt bath which makes it different from radiantly-heated furnaces holds a six-way advantage for the automotive industry.

The characteristics:

1. All work immersed in salt is completely sealed from air so that scaling is impossible.

2. Effect of the work on the surface can be chemically controlled.

3. While capable of extremely rapid heating (about one-fourth the normal time), it cannot overheat and the heat shock on loading into the molten salt is reduced. Salts are available to permit operation from 300 to 2400 F.

The advantages:

1. The salt bath's high heating rate permits large production per unit of floor space.

2. Installed cost per unit of production is low, often the lowest available.

3. It provides accurate temperature control. Not only can the furnace be held to ± 5 deg in any portion of the bath, but it is impossible to overheat the work, no matter how long it may be soaked at furnace temperature.

4. It provides complete surface protection without special apparatus.

5. It drastically reduces heat-treatment distortions. Savings in finishing operations greater than total heat-treatment costs are possible.

6. Salt bath furnaces can be completely mechanized.

Major applications of the salt bath have been in cyaniding, carburizing, neutral hardening, austempering, martempering, cyclic annealing, cleaning, desanding, descaling, and bluing.

* Based on "Recent Developments in Heat-Treating Processes," presented at SAE National Passenger Car, Body, and Production Meeting, Detroit, March 9, 1949. (This material is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

SAE National

TRACTOR Meeting

Hotel Schroeder
Milwaukee, Wis.

Sept. 13-15

TUESDAY, Sept. 13

9:30 a.m.

Welcome to Milwaukee
—G. J. HAISLMAIER, Chairman, SAE Milwaukee Section

J. W. Bridwell, Chairman

Symposium—Utilization of Horsepower in Earthmoving Equipment

Use of Large-Capacity Slow-Speed Equipment
—J. E. MARSON, Bucyrus-Erie Co.

Obtaining Higher Average Speed
—J. P. CARROLL, Caterpillar Tractor Co.

Use of Higher Maximum Speeds
—H. L. RITTENHOUSE, The Euclid Road Machinery Co.

(Sponsored by Construction and Industrial Machinery Subcommittee)

12:00 noon Fifth Floor—East Room
Special Luncheon

1:30 p.m.

R. A. Beckwith, Chairman

Hydraulic Power as Applied to Earthmoving Equipment
—E. J. HRDLICKA, JR., Hydraulic Equipment Co.

Research on the Performance of Power Cranes and Shovels
—TREVOR DAVIDSON and J. H. MEIER, Bucyrus-Erie Co.

(Sponsored by Construction and Industrial Machinery Subcommittee)

WEDNESDAY, Sept. 14

9:30 a.m.

H. L. Brock, Chairman

Dust and Its Effect on Air Cleaner Design
—W. W. LOWTHER, Donaldson Co., Inc.

Lubricant Retention and Dirt Exclusion on Farm Implements
—S. C. McCOMBS and R. O. ISENBARGER, Chicago Rawhide Mfg. Co.

(Sponsored by Tractor and Farm Machinery Activity)

12:00 noon Fifth Floor—East Room
Special Luncheon

1:30 p.m.

W. H. Worthington, Chairman

Symposium — Hardening Gear Teeth by Induction Heating

High-Frequency Heat Treatment of Gears—Equipment and Processes
—J. A. REDMOND, Westinghouse Electric Corp.

Induction Hardened Gears
—H. B. KNOWLTON and H. F. KINCAID, International Harvester Co.

(Sponsored by Tractor and Farm Machinery Activity)

THURSDAY, Sept. 15

9:30 a.m.

D. C. Heitshu, Chairman

Tillage Forces and Their Effect on the Farm Tractor
—A. W. CLYDE, Pennsylvania State College

Gear Drives in Implement Design
—E. E. EATON, Clark Equipment Co.

(Sponsored by Farm Implement Subcommittee)

12:00 noon Fifth Floor—East Room
Special Luncheon

1:30 p.m.

W. E. Knapp, Chairman

New Cultivating Methods
—S. D. POOL, International Harvester Co.

Soil Conservation Practices as Related to Farm Tractor and Implement Design
—G. E. RYERSON, Soil Conservation Service, U. S. Department of Agriculture

(Sponsored by Farm Implement Subcommittee)

DINNER

7:00 p.m. THURSDAY

L. A. Gilmer, Chairman
C. E. Frudden, Toastmaster
S. W. Sparrow, SAE President

"Progress in Rubber"
John L. Collyer, President
B. F. Goodrich Co.

FRIDAY, Sept. 16

10:00 a.m. Location to be announced
SAE Milwaukee Section Annual Golf Tournament
Dinner—6:30 p.m.
Awarding of prizes 8:00 p.m.

Hydrodynamics of the Hydraulic Torque Converter

EXCERPT FROM PAPER* BY **Ernst W. Spannhake** CONSULTING ENGINEER

(The complete paper from which this excerpt is taken will be printed in full in SAE Quarterly Transactions. It discusses also certain refinements of torque converter theory and the differences between torque converters and other hydraulic machinery.)

THERE are many torque converters with widely varying characteristics in the field at the present time. Since the object of most of them is to replace a multiple-step gearbox as fully as possible, it seems appropriate that some means should be found for comparing torque converters of various performance characteristics as to their relative merit.

Fig. 1 shows the performance characteristics of a typical low-speed, high-torque converter. T_s/T_p signifies the torque ratio; η, the efficiency; $n_s/n_p = i$, the speed ratio between output and input shaft; and N_p the primary load for a constant speed. As highest speed ratio, $i = 0.4$ is chosen.

Fig. 2 on the other hand, gives a characteristic picture of a typical high-speed, low-torque converter with the same lettering signifying the same magnitudes. Here the highest speed ratio is $i = 0.8$. By themselves and as single units, it is obvious that these two converters give entirely different performance. We must remember, however, that converters are used like any other transmission to drive machinery. That is, from the output shaft of the torque converter there will be some mechanical transmission or gear train leading to the final drive shaft of the vehicle, earthmoving machinery, or any other application. The size and transmission ratio of this gear train will be chosen in such a way that its highest operating speed will correspond to the highest operating speed of the converter shaft. The performance of the entire machinery in question will be determined by the torque-speed characteristics of this shaft. A short consideration will show that the widely different torque converters, as shown in Figs. 1 and 2, can be made to give essentially the same performance. This is done in Fig. 3, where for

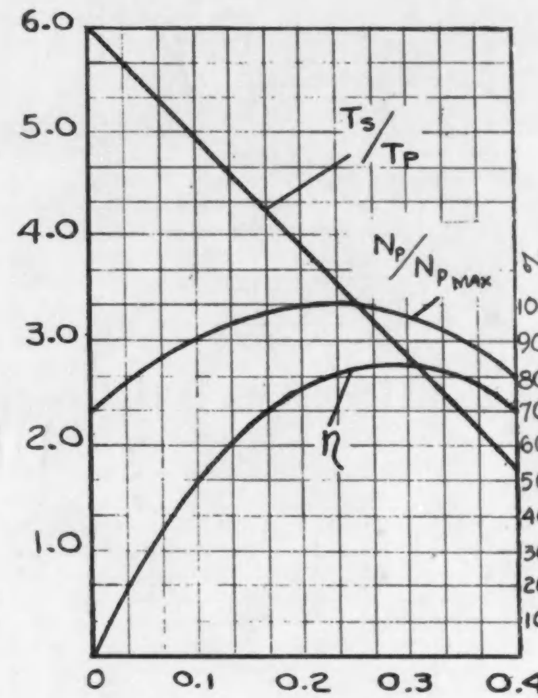


Fig. 1—Slow-speed, high-torque converter

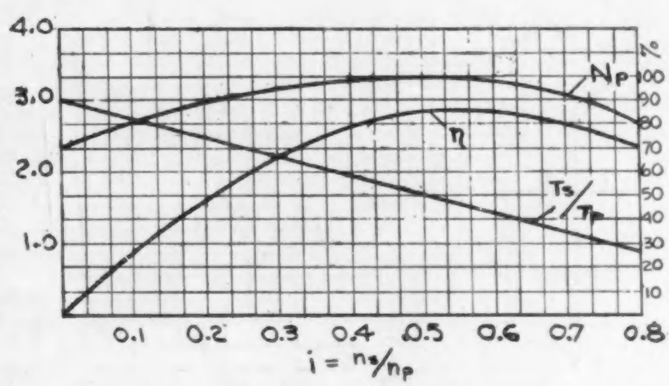


Fig. 2—High-speed, low-torque converter

*Paper "Hydrodynamics of the Hydraulic Torque Converter" was presented at SAE Summer Meeting, French Lick, Ind., June 6, 1949. (This paper is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

the sake of convenience, the highest rpm of the machine and driveshaft has been assumed to be one quarter of the engine shaft rpm.

In this picture:

T_E = Engine speed
T_O = Operating torque
n_O = Operating rpm
η_O = Operating efficiency

The operating efficiency is the torque converter efficiency multiplied by the gear train efficiency, which can reasonably be assumed to be essentially independent of its overall speed ratio.

In these figures, the characteristics of the two torque converters were chosen to give an exactly equivalent performance when coupled with the gear train. The reason for the equivalent performance of the two converters is, of course, founded on the fact that in this particular picture great pains have been taken to draw each curve in such a manner that for equal fractions of the maximum speed ratio, the torque ratios as multiples of the torque ratio at this maximum speed — and also the efficiencies and primary loads — have the same numerical value.

It seems therefore highly advantageous for the purpose of comparison to plot the performance characteristics of different torque converters in the shape of a unit diagram, preferably in such a way that torque ratio as a fraction of the highest operable one, efficiency, and primary load are plotted against speed ratio as a fraction of the highest operable one. In Fig. 4 such a unit diagram is shown. Since for most applications, an efficiency of less than 70% for the torque converter alone will mark the upper limit of operable speeds, a good definition of maximum speed ratio would be the

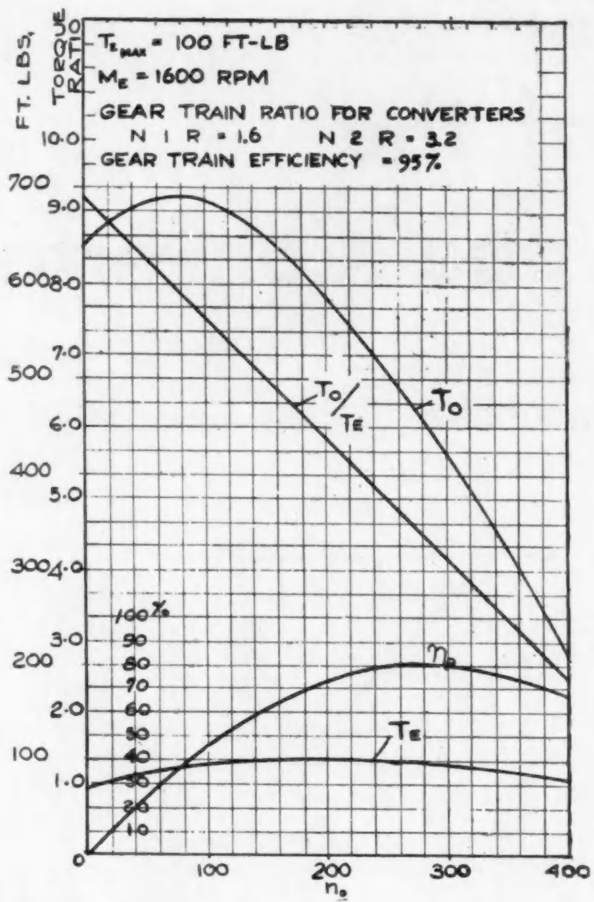


Fig. 3—Converters of Figs. 1 and 2 driven by constant-speed engine

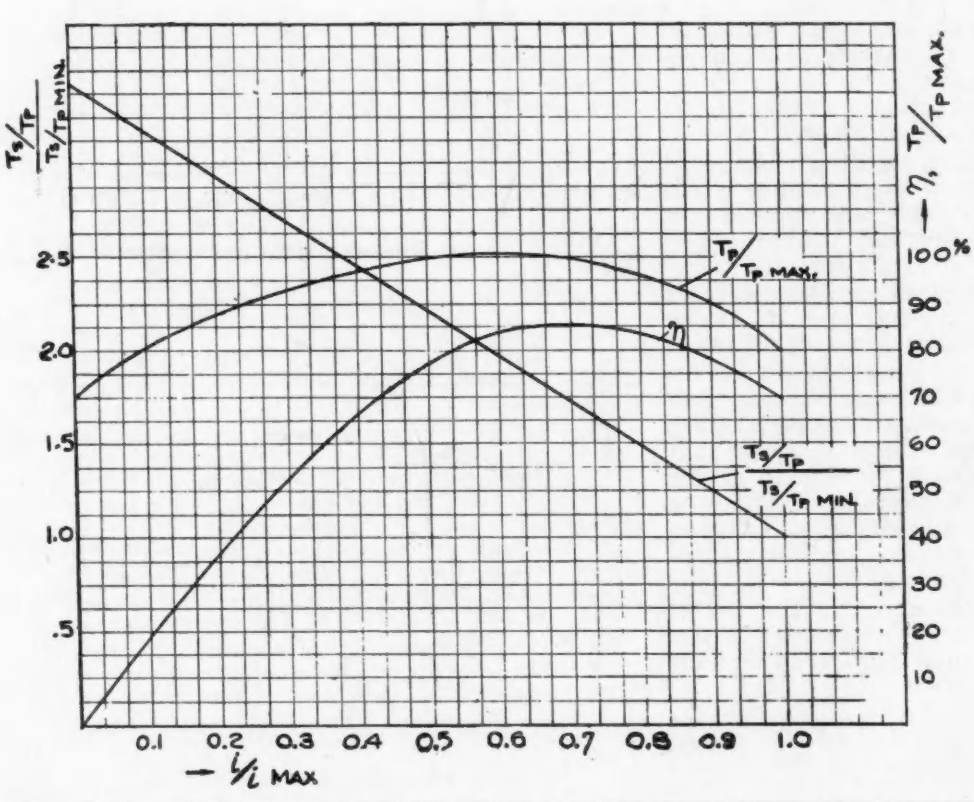


Fig. 4—Unit diagram of converters shown in Figs. 1 and 2

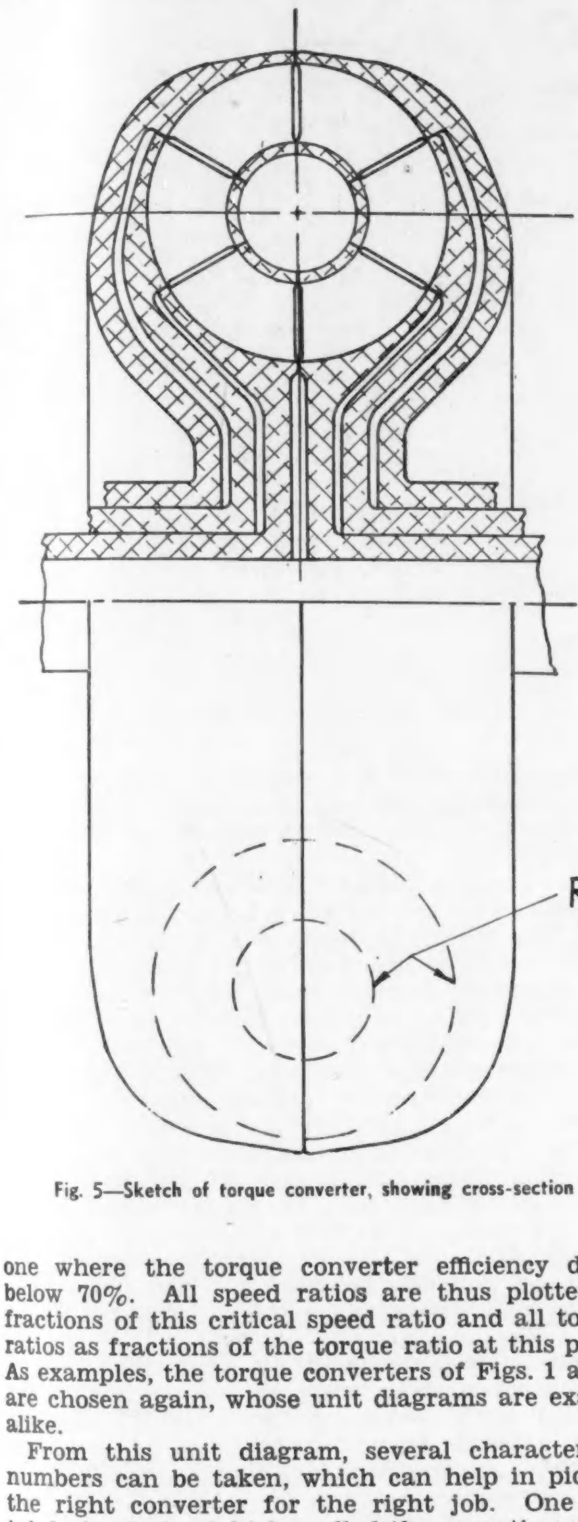


Fig. 5—Sketch of torque converter, showing cross-section

one where the torque converter efficiency drops below 70%. All speed ratios are thus plotted as fractions of this critical speed ratio and all torque ratios as fractions of the torque ratio at this point. As examples, the torque converters of Figs. 1 and 2 are chosen again, whose unit diagrams are exactly alike.

From this unit diagram, several characteristic numbers can be taken, which can help in picking the right converter for the right job. One certainly is what might be called the operating range —in other words, the range of speed ratios in which the efficiency stays above a minimum tolerable efficiency, in this case 70%. Another is the effective stalling ratio; this means the operating stalling torque ratio divided by the torque ratio at the maximum operating speed. Another ratio which should be interesting is the primary torque ratio, which will be given as fraction of the maximum primary torque at operating speed. One could also establish what might be called the overspeed ratio. That is that speed which can be achieved (of course at rather poor efficiencies) until the output torque goes to zero, at which point, of course, the efficiency is zero also.

From the above discussion, it is apparent that a wide variety of torque converters can be fitted into the unit diagram. It is interesting to note that present-day torque converters do not differ too widely from each other when plotted in this fashion.

If torque converters are regarded in view of their performance according to the unit diagram, it is much easier to obtain a picture of the relative merits of any change in the shape of their principal design data, and for this reason the author will try in this paper to follow this kind of analysis.

Characteristics of Converters

In analyzing the hydrodynamic characteristics of the torque converters, it should of course be kept in mind that basically the torque converter is nothing but a succession of well-known types of hydrodynamic wheels, turbine wheels, pump wheels, and guide-vane rings, as they occur in any hydraulic machinery.

Fig. 5 gives a sketch of such an agglomeration. No attempt is made here to connect any of the wheels shown to primary, secondary, or stationary shafts, this being unessential for showing the principle of a hydrodynamic circuit. The principal difference between ordinary hydraulic machinery and torque converter wheels is that in the converter they are arranged in such a way that the operating fluid flows through them in succession, one after the other, so that after it has passed the last wheel, it enters into the first one again. A lot of the basic differences and basic difficulties of torque converter design spring from this fact, which might be described in a few words by saying that the hydrodynamic wheels must form a hydrodynamic circuit.

Disregarding the circumferential components which each wheel imposes upon the fluid, it can therefore be seen that the basic fluid motion in the torque converter is the continuous motion in an annular ring, which in the lower part of Fig. 5 is shown designated by R. Two important characteristics of the torque converter circuit can, therefore, be derived immediately from this picture:

1. In each torque converter the amount of fluid in pounds per second passing through each wheel is the same.
2. The annular ring structure of any torque converter is charged basically with the problem of turning the direction of the flow around by 360 deg in going once around its periodical path.

From the first condition immediately follows the basic law of torque conversion in a converter. According to Euler's principle, the torque which is the mechanical torque exerted upon the wheel by the fluid is directly proportional to the difference of moment of momentum of the fluid entering and leaving the wheel. The moment of momentum M however can be written:

$$M = Q \times (c_u \cdot r)_{mean} \qquad (1)$$

Here Q is the discharge through the wheel in slugs per second, c_u is the circumferential component of a fluid particle velocity in a space between

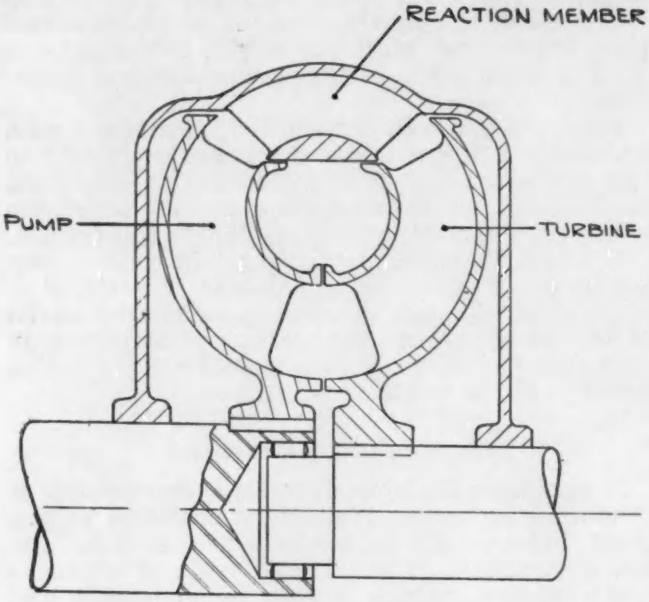


Fig. 6—Diagram of torque converter

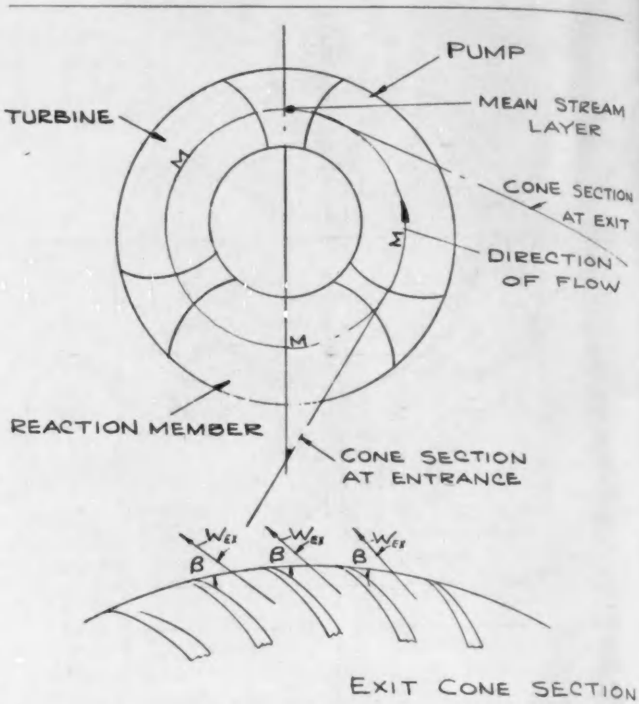


Fig. 7—Sketch showing cone sections

wheels, and r the radius at that point.

Of course the way it is written is nothing new over the previous statement and simply would be a definition of $c_u \cdot r_{mean}$. However, it is a fact that in the space between wheels any flow tends to assume the configuration in what is called a free vortex, which is characterized by the condition that everywhere within this space $c_u \cdot r$ tends to be constant. Therefore, $(c_u \cdot r)_{mean}$ tends to be not much different from $c_u \cdot r$ at any point in the space between wheels, and therefore has a real meaning. We shall call it the vortex value.

Since now the various torques of all wheels are made up of the difference in the vortex values between exit and entrance and since furthermore in going through every wheel the fluid has to come back to its point of origin again, it can be seen that the sum total of all differences has to be zero, which of course means that the sum of all torques has to be zero.

This is, of course, nothing but the old law of the conservation of torque, but for the designer of torque converters, it is interesting and sometimes valuable to visualize hydrodynamically how this is happening in the particular machinery. It simply means that any increase in vortex value somewhere in the torque converter has to be decreased again somewhere else and vice versa. An interesting example of this is the attempt in a torque converter, like for example that shown in Fig. 6, where the exit of the turbine is followed by the entrance into the pump in the direction of flow.

If insufficient torque ratio is experienced in such a design, for instance at stalling, it is only too natural to assume that by bending the turbine wheels still further backward, thus creating a larger backward vortex value, a larger turbine torque and therefore greater torque ratio will be achieved. On closer examination, it is seen however that this means that now also the pump will receive the fluid at a higher backward vortex value. Its momentum difference will, therefore, be increased by the same amount as that of the turbine so that the torque ratio, assuming that losses and discharge rate would stay the same, will actually be decreased by such a measure.

This simple consideration shows us already that in estimating the influence of one single change anywhere in the hydraulic circuit usually the circuit as a whole and not only the wheel immediately affected has to be considered. However in our foregoing example we have already made one additional assumption which is not usually correct, that the discharge rate stays unaffected. Usually any change in the converter effects the discharge rate, and, as we know, all hydrodynamic wheels and especially the moving ones change their behavior considerably with varying discharge. Any attempt to evaluate effects of changes in a torque converter should therefore take into account the effects of the varying discharge rate. Our problem therefore is solved if we can find how the vortex values in the gaps and the discharge rate are influenced by the design data of the converter.

Fortunately, some simplifying assumptions can be made which are reasonably accurate, especially for the kind of torque converters we are dealing with, which are characterized by rather narrow spacing of blades. For these the following assumptions hold true approximately:

1. The law of similarity holds; that is, the losses are independent of Reynolds number.

2. The vortex value is influenced only by the discharge and the blades of the wheel preceding it.

3. The blades exert a certain guiding effect upon the fluid, making it leave the wheel relative to itself in approximately the same direction, regardless of speed and discharge.

4. The vortex value created by such a wheel can be calculated as being the one which would be

created by a ***mean characteristic portion of the blade.***

None of these assumptions is quite correct—the first being better than the last—but they give rise to fairly simple mathematical means for quickly determining whether a converter is basically low speed or high speed, what its tendency for high efficiency ranges is with all other things being equal, and other important data. A short mathematical analysis of the type of torque converter desired on this basis could therefore always be of interest, especially since at least in basic characteristics, experience seems to confirm results of this analysis.

According to Assumption 4, we regard as the characteristic section of the torque converter a certain central portion of the annular ring space as indicated by M in Fig. 7, which we call the mean streamlayer and observe what is happening in it.

Since for our considerations only the exit and entrance of the wheels are important, it is best to use the time-tried method for all hydraulic machinery and take a cone section tangential to the streamlayer at the point of entrance and exit to the wheel as indicated in Fig. 7, which gives a symbolic picture of a hydrodynamic circuit.

The velocity relations obtained from this picture at the exit are the same as for all hydraulic wheels and consist of the relations

$$\bar{c} = \bar{u} + \bar{w} \qquad (2)$$

where $\bar{u}$ signifies the circumferential speed of the wheel at the blade tip, $\bar{w}$ the relative velocity of the fluid in regard to the blades, and $\bar{c}$ the absolute velocity of the fluid coming out of the wheel. The addition is, of course, meant to be taken in a geometric or vectorial sense. A vectorial diagram of this relation is shown in Fig. 8.

According to Assumption 3 in this picture, the wheel will give the fluid always the same directional angle β. The velocity $\bar{w}$, of course, can vary. It is determined by the quantity c_m, the annular velocity, which determines the quantity flowing through the converter.

This gives rise to the relation

$$c_u = u + w_u \qquad (3)$$

Here the positive direction is in the same direction as the rotation of the wheel, and c_u and w_u are the circumferential components of $\bar{c}$ and $\bar{w}$ respectively.

At the entrance to the wheel, a similar relation holds, as can also be seen in Fig. 8. Here we have however a slight difference. According to our Assumption 2, the components c_u and therefore $\bar{c}$ is not influenced by the blades they meet. In the free space before meeting the blades, the relative velocity in regard to the wheel is $\bar{w}_f$ as shown in the picture. Except for one set of conditions, the blade entrance will not be built to receive this flow smoothly. Since most of the falling off in efficiency in torque converters comes from this condition, a certain simplified approach for analysis is necessary. This has been found in the assumption that the wheels immediately after the flow enters "shock" the flow by their guiding effect into a direction which it should have had for shockless entrance. The loss—at least for fairly narrowly spaced blades as used in most converters—can then be brought into relation to the "shock" velocity w_{sh} which is necessary to bring about the change from $\bar{w}_f$ to w_{wh}. This velocity in

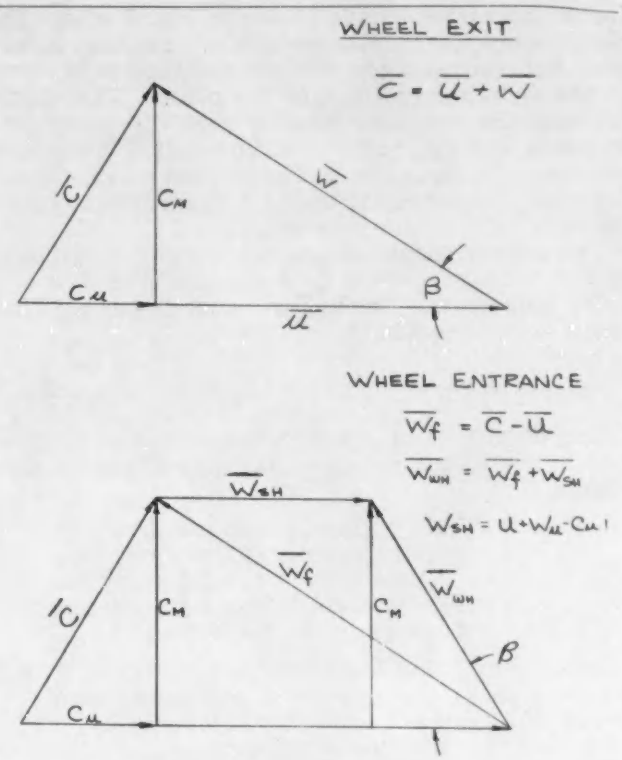


Fig. 8—Velocity diagrams for wheel

our case becomes

$$w_{sh} = u + w_u - c_u \qquad (4)$$

We now regard our torque converter system in Fig. 7. It is obvious that if the pump starts to rotate, the fluid inside starts to be pumped in the direction indicated by the pump design. Its velocity will continue to increase in magnitude until all energy put into the fluid by the pump is consumed by either the mechanical energy which is taken out of the fluid by the turbine, or by hydraulic losses.

For a simple estimation of the hydraulic losses, we can now make use of our Assumption 1. This supposes that all characteristics will remain the same if we leave the discharge and speed ratio $i = \frac{n_s}{n_p}$ (n_p being the primary rpm and n_s the secondary rpm) constant and vary the pump speed until the energy balances. This has the advantage that in our velocity diagrams of Fig. 8 c_m, the annular speed component, and therefore the relative velocity $\bar{w}$, stays constant, which makes the calculations easier.

Now again using our similarity assumption, we reduce every magnitude to unit figures by assuming the radius at the pump exit to be unity and the annular velocity c_m there to be unity also. Then all relative speeds $\bar{w}$ within the wheels are fixed. If we now call the circumferential speed at the exit of the pump α, the angular speed w is also equal to α, the radius at that point having been set equal to unity.

We have now all the means for calculating the discharge characteristics of the torque converter if we can express all velocities, energies, and losses in terms of α, the speed ratio i, and the blade angles, which are determined in terms of w_u for a unit c_m.

As an example, we give the solution for a simple

torque converter circuit shown in Fig. 9 where the circuit sequence is pump—turbine—reaction member. Exit radius of one wheel is assumed to be equal to the entrance radius into the other. The radius between the reaction member and the pump entrance is called ϱ_p, between turbine exit and reaction member ϱ_s, whereas as said before the exit radius of the pump, being equal to the entrance radius into the turbine, is assumed to be unity.

The energy exchanged between moving turbohydraulic wheels and the operating fluid that is the equivalent of the mechanical energy put into or taken out of the fluid is

$$H = \frac{\omega}{g}\left[(c_u \cdot r)_{exit} - (c_u \cdot r)_{entr}\right] \qquad (5)$$

which, as in case of the torque, relates this magnitude to the vortex values in the spaces between the wheels.

Here H = Fluid energy in ft-lb per lb
ω = Angular velocity of the wheel in radians per sec
g = Gravity acceleration in ft per sec^2
$c_u \cdot r$ = Vortex value in ft^2 per sec

This vortex value is positive for energy put into the fluid as in the case of a pump, negative for energy taken out as in the case of a turbine.

We now use our similarity symbols for constant unit c_m at the pump exit. For this condition we give to the circumferential relative velocity components imposed upon the fluid by the wheels the following notation:

Pump:	entrance	$w_u = w_{11}$
	exit	$w_u = w_{12}$
Turbine:	entrance	$w_u = w_{21}$
	exit	$w_u = w_{22}$
Reaction member:	entrance	$w_u = w_{31}$
	exit	$w_u = w_{32}$

The blade tip velocities become with the notations as indicated before:

$$\left.\begin{array}{lll} \text{Pump:} & \text{entrance} & u = \alpha \cdot \varrho_p \\ & \text{exit} & u = \alpha \\ \text{Turbine:} & \text{entrance} & u = i \cdot \alpha \\ & \text{exit} & u = i \cdot \alpha \cdot \varrho_s \\ \text{Reaction member:} & \text{entrance} & u = 0 \\ & \text{exit} & u = 0 \end{array}\right\} \qquad (6)$$

With this and using Eq. (3), we can write the vortex values in the spaces I, II, and III shown in Fig. 9.

$$\left.\begin{array}{ll} \text{Guide Vane–Pump} & \\ (c_u)_I = w_{32} & (c_u \cdot r)_I = w_{32} \cdot \varrho_p \\ \text{Pump–Turbine} & \\ (c_u)_{II} = \alpha + w_{12} & (c_u \cdot r)_{II} = \alpha + w_{12} \\ \text{Turbine–Reaction Member} & \\ (c_u)_{III} = \alpha \cdot i \cdot \varrho_s + w_{22} & (c_u \cdot r)_{III} = \alpha \cdot i \cdot \varrho_s^2 + w_{22}\varrho_s \end{array}\right\} \qquad (7)$$

With this notation, our mechanical energies exchanged into fluid energies become:

For the pump:

$$H_p = \alpha(\alpha + w_{12} - w_{32}\varrho_p) \qquad (8)$$

For the turbine:

$$\begin{aligned} H_t &= -i\alpha(\alpha + w_{12} - \alpha \cdot i\varrho_s^2 - w_{22}\varrho_s) \\ &= -i\alpha[\alpha(1 - i\varrho_s^2) + w_{12} - w_{22}\varrho_s] \end{aligned} \qquad (9)$$

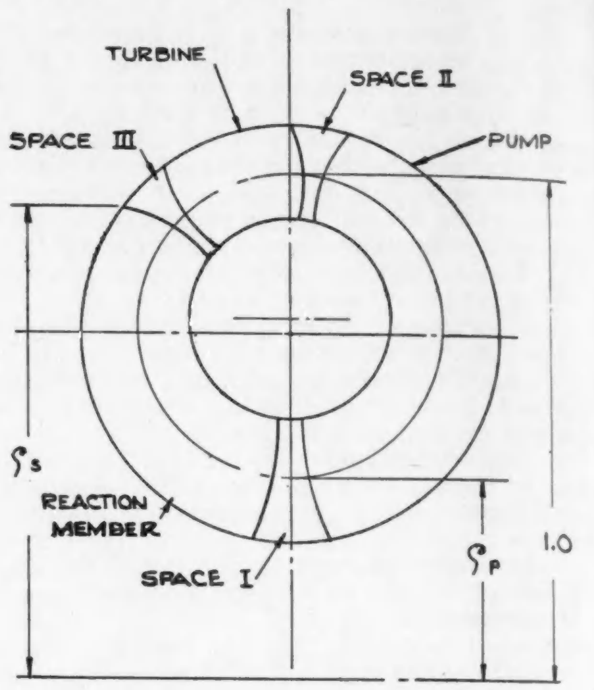


Fig. 9—Sketch explaining symbols

For a first approximation of the shock losses, since the earliest inception of the torque converter, an expression like

$$H_s = c_{sh}\frac{w_{sh}^2}{2g} \qquad (10)$$

has been found useful, c_{sh} being a coefficient depending upon such factors as entrance conditions and head shapes. For research calculations a value of $c_{sh} = 1.0$ has been found to give in many cases satisfactory approximation of torque converter behavior.

The shock velocities in unit notation using Eqs. (4) and (7) can be written:

$$\left.\begin{array}{ll} \text{Pump entrance:} & (w_{sh})_I = \alpha \cdot \varrho_p + w_{11} - w_{32} \\ \text{Turbine entrance:} & (w_{sh})_{II} = i \cdot \alpha + w_{21} - \alpha - w_{12} \\ \text{Reaction member} & \\ \quad\text{entrance:} & (w_{sh})_{III} = w_{31} - \alpha \cdot i\varrho_s - w_{22} \end{array}\right\} \qquad (11)$$

Whereas these shock losses can be more or less reduced by careful design of blade shapes, the loss due to fluid friction will always be present. With not too extreme blade shapes it can be set equal to

$$H_f = \lambda \cdot \frac{w_{ex}^2}{2g} \qquad (12)$$

for each wheel, if as is usually the case the relative exit velocity w_{ex} is the highest relative velocity occuring in the wheel. The coefficients, which are similar in nature to pipe friction coefficients, vary with the Reynolds Number, the roughness of the surface, and the configuration and length of the fluid passages. Low values of 0.08 have been observed; $\lambda = 0.25$ seems high; 0.15 is probably attainable in larger converters after careful development.

Since according to Fig. 8 we can set

$$w_{ex}^2 = w_u^2 + c_m^2$$

we obtain

$$\left.\begin{aligned} &\text{Pump:} && H_f = \frac{\lambda}{2g}(w_{12}^2 + 1) \\ &\text{Turbine:} && H_f = \frac{\lambda}{2g}[w_{22}^2 - (c_m^2)_{turb}] \\ &\text{Reaction member:} && H_f = \frac{\lambda}{2g}[w_{32}^2 - (c_m^2)_{react}] \end{aligned}\right\} \qquad (13)$$

The values for c_m at turbine and reaction member exit in many converter designs are being made equal to 1.0. In this case:

$$H_f = \frac{\lambda}{2g}(3 + w_{12}^2 + w_{22}^2 + w_{32}^2) \qquad (14)$$

Since we assume for our balance equation all w's to be constant, the total friction loss, H_f, assumes a constant value which we call $\frac{L}{g}$.

Our energy balance equation then assumes the simple shape:

$$H_p + H_t = \Sigma H_L \qquad (15)$$

Here H_p and H_t are taken with their proper sign, and ΣH_L is the sum of all hydraulic losses.

Figuring only shock losses and friction losses, which can be done to obtain an overall picture, we obtain as balance equation:

$$\overbrace{\frac{\alpha}{g}(\alpha + w_{12} - w_{32}\varrho_p)}^{H_p} + \overbrace{\frac{i \cdot \alpha}{g}(\alpha \cdot i\varrho_s^2 + w_{22}\varrho_s - \alpha - w_{12})}^{H_t}$$
$$= \frac{1}{2g}[\alpha \cdot \varrho_p + (w_{11} - w_{32}]^2\} H_{s_{pump}}$$
$$+ \frac{1}{2g}[\alpha(1 - i) - (w_{21} - w_{12})]^2\} H_{s_{turb}} \qquad (16)$$
$$+ \frac{1}{2g}[\alpha i\varrho_s + (w_{22} - w_{31})]^2\} H_{s_{react}}$$
$$+ \frac{L}{g}\} \Sigma H_f$$

This is a quadratic equation which can be solved easily for α:

$$\begin{aligned} \alpha^2[(1 - \varrho_p^2) - i^2(1 - \varrho_s^2)] \\ - 2\alpha[w_{11}\varrho_p - w_{21}) - i(w_{21} - w_{31}\varrho_s)] \\ - [(w_{22} - w_{31})^2 + (w_{12} - w_{21})^2 \\ + (w_{32} - w_{11})^2] + 2L) = 0 \end{aligned} \qquad (17)$$

Solving this equation for each i, then, gives the whole characteristic of the torque converter by reconverting everything to constant primary speed and using the law of similarity for turbohydraulic wheels.

Thus the torque ratio:

$$T_R = T_t/T_p = \frac{Q[(c_u \cdot r)_{II} - (c_u \cdot r)_{III}]}{Q[(c_u \cdot r)_{II} - (c_u \cdot r)_I]}$$
$$= \frac{\alpha(1 - i\varrho_s^2) + w_{12} - w_{22}\varrho_s}{\alpha + w_{12} - w_{32}\varrho_p} \qquad (18)$$

The efficiency:

$$\eta = i\frac{T_r}{T_p} = \frac{i[\alpha(1 - i\varrho_s^2) + w_{12} - w_{22}\varrho_s]}{\alpha + w_{12} - w_{32}\varrho_p} \qquad (19)$$

The annular velocity (for constant primary speed):

$$c_m = \frac{1}{\alpha} \times u_p$$

Here u_p = exit blade tip speed of pump.

For the pump torque it is essential to remember that, just as for a typical pump, it goes through a

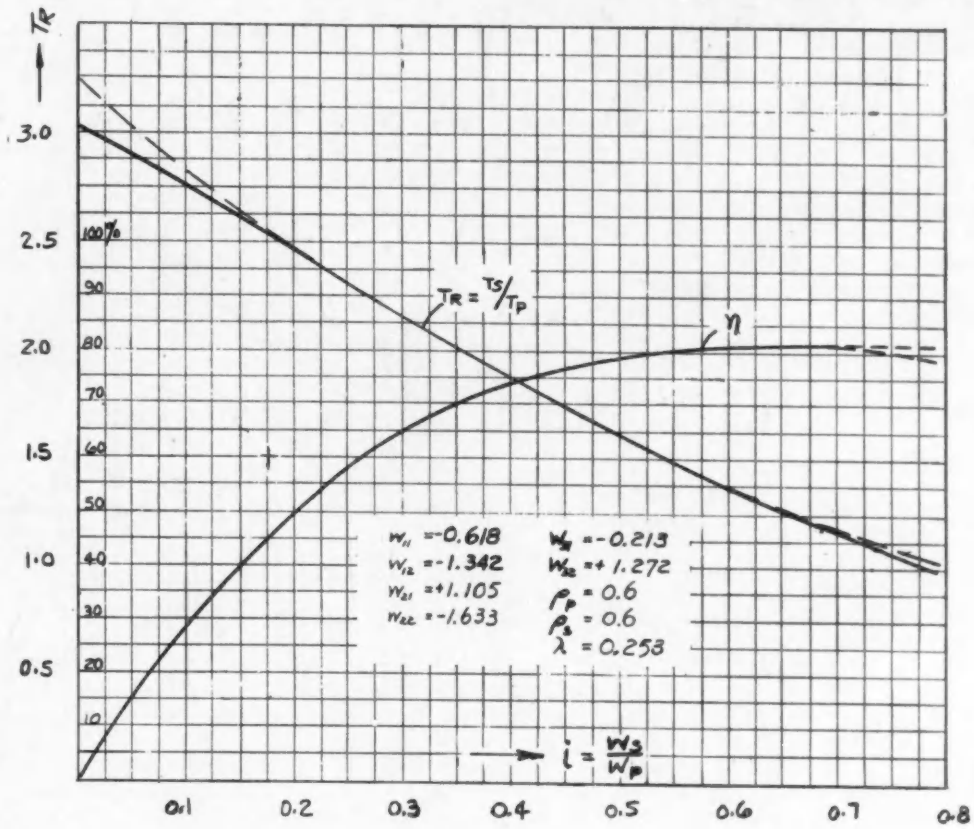


Fig. 10—Comparison between calculated and test results of performance of experimental converter

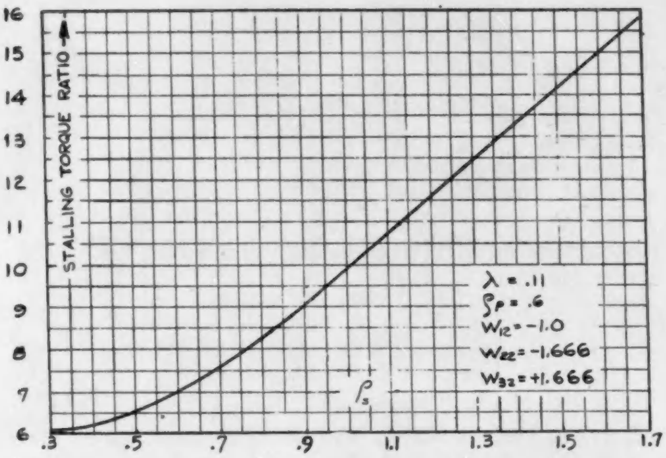


Fig. 11—Stalling torque ratio as a function of turbine exit radius. Optimum shock losses

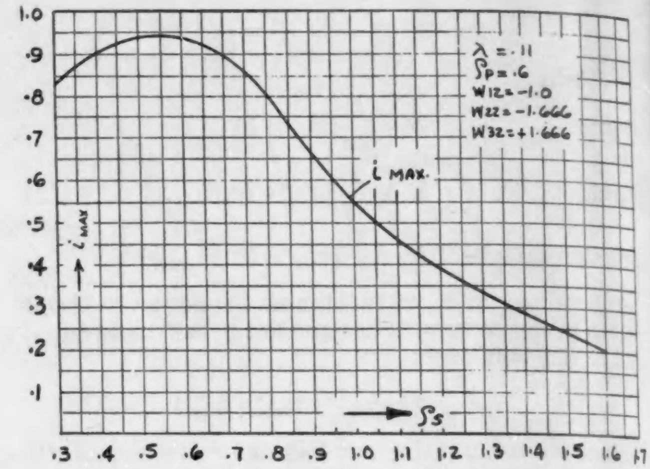


Fig. 12—Maximum operable speed ratio as a function of turbine exit radius. Optimum shock losses

maximum with discharge. In our case—as can easily be verified by calculation—this maximum will be at:

$$\alpha = -2(w_{12} - w_{32}\varrho_p) \tag{20}$$

and its value will be

$$T_{p_{max}} = \frac{-1}{4(w_{12} - w_{32}\varrho_p)} \tag{21}$$

As a fraction of this maximum torque, the primary torque becomes:

$$t_p = \frac{T_p}{T_{p_{max}}} = \frac{-4(w_{12} - w_{32}\varrho_p)(\alpha + w_{12} - w_{32}\varrho_p)}{\alpha^2} \tag{22}$$

As a further illustration of the type of research which can be accomplished by means of this method, Figs. 11 and 12 present maximum speed ratio and stalling torque as a function of turbine exit radius for a three-wheel torque converter. Large radii favor stalling torque, small radii the maximum speed ratio. Which of the two is better can be seen only in the unit diagrams. These are given in Fig. 13. Here the smaller radii are definitely favored for overall performance. Other considerations of the influence of design changes are also solved in principle in the paper by means of these equations.

These formulas enable one to determine fairly accurately the behavior of a converter type. As an example, there is given in Fig. 10 the comparison of the performance of a torque converter—which was tested some years ago and happened to be a passing stage in a certain development—with the calculation of its performance by the above equations.

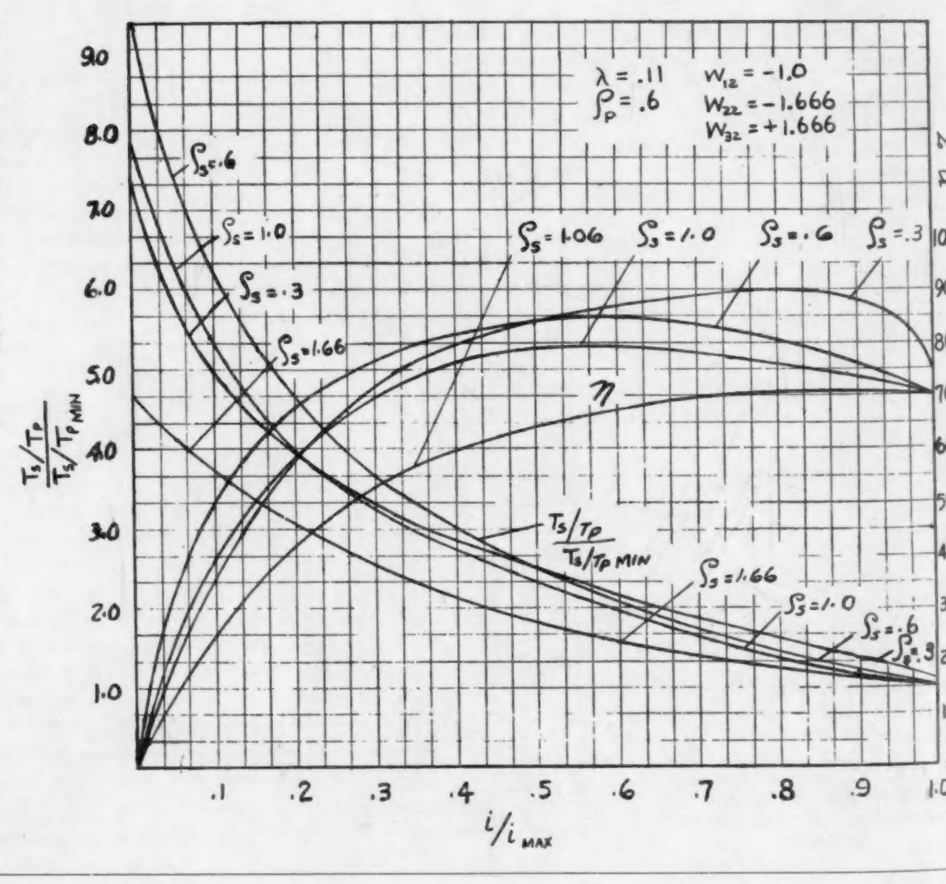


Fig. 13—Unit diagram of converters described in Figs. 11 and 12. Optimum shock loss

TABLE I

IRON BASE HIGH TEMPERATURE ALLOYS

Alloy	Timken	19-9DL	GT-45	N-153	S-588	N-155 Multimet	S-590	G-18-B
C	0.15	0.26	0.08	0.10	0.42	0.10	0.49	.40
Mn	1.14	0.52	1.25	1.50	1.50	1.54	0.60	.80
Si	0.84	0.57	0.50	0.50	0.80	0.44	0.21	.90
Cr	16.75	18.95	17.30	17.00	18.38	21.44	19.50	12.80
Ni	25.62	9.05	13.80	15.00	20.31	20.40	19.78	13.26
Co	....	....	....	13.00	.....	19.82	19.35	10.42
Mo	6.29	1.22	2.90	3.00	4.18	2.96	3.95	1.62
W	...	1.19	...	2.00	3.08	2.11	4.15	2.89
Cb	...	0.30	0.45	1.00	4.69	1.08	4.04	3.23
Ti	...	0.21	0.30	...	...	...	...	...
N	0.13	...	...	0.10	...	0.13	...	...
Cu	...	...	3.10	...	...	...	...	...
Fe	Bal.	Bal.	Bal.	Bal.	Bal.	Bal.	Bal.	Bal.

Jets, Turbines Demand New Forging Methods

Based on paper by

L. S. FULTON

Universal Cyclops Steel Corp.

TABLE II

NICKEL BASE HIGH TEMPERATURE ALLOYS

Alloy	Hastelloy B	Hastelloy C	Refractaloy 26	Nimonic 80
C	0.06	0.06	0.03	0.04
Mn	0.60	0.70	0.70	0.56
Si	0.40	0.40	0.65	0.47
Cr	...	15.00	17.90	21.18
Co	...	....	20.00	....
Mo	28.00	17.00	3.03	....
W	...	5.00	...	....
Ti	...	...	2.99	2.44
Al	...	...	0.25	0.63
Fe	5.00	5.00	19.00	1.00
Ni	Bal.	Bal.	Bal.	Bal.

TABLE III

COBALT BASE HIGH TEMPERATURE ALLOYS

Alloy	S-816	HS #21 Vitallium	HS #23 61	HS #27 6059	HS #30 422-19	HS #31 X40
C	0.36	0.30	0.42	0.40	0.40	0.48
Mn	0.40	0.35	0.28	0.30	0.30	0.64
Si	0.28	0.55	0.57	0.65	0.51	0.72
Cr	20.24	28.70	24.21	24.61	24.75	25.12
Ni	20.51	2.00	1.00	33.00	15.92	9.69
Mo	3.85	5.57	...	5.34	6.08	...
W	4.39	...	5.38	...	...	7.23
Cb	3.92	...	...	...	...	...
Fe	3.28	1.00	1.00	1.00	1.00	1.00
Co	Bal.	Bal.	Bal.	Bal.	Bal.	Bal.

FORGE shop technicians alone can overcome difficulties in producing turbine buckets, by far the most difficult manufacturing problem posed by jet and turbine powerplants.

Currently used iron, nickel, and cobalt base alloys, Tables I, II, and III, are relatively new typical materials with which there has been little experience. For example, only S-816 and vitallium (HS-23) of the cobalt base alloys have been forged successfully to date.

The usual practice in supplying billets is to melt in an ingot mold that will give at least a reduction of three to one from ingot to billet. Ingots are then either preheated or charged into a cool furnace and heated to 1900 to 2150 F in a neutral or slightly oxidizing atmosphere.

First press or hammer operations are very light until the metal begins to flow. Then reductions can be increased, but with caution needed for all highly alloyed materials.

Forging is stopped on most such alloys at about 1800 F or slightly less, and the semiforged part is again reheated. From four reheatings of the lower alloy content to eight or more on such alloys as S-816 are required.

All billets are ground two or three times and stress relieved, but smaller bars are given from four to five grindings, rough turning, or centerless grinding operations to produce the finished size.

Root sections of the turbine buckets are either upset or extruded. They are then forged in repeated operations under small hammers. Forging temperatures are usually lost after two or three blows of the hammer, and they must again be reheated. Besides the S-816, numerous buckets are being forged from such high alloys as Timken, 19-9DL, N-155, Inconel X, and Hastelloy B and C. All materials are furnished centerless round.

When the forgings are finished they are inspected by Magnaflux, Zyglo, reflectoscope, and X-ray for inherent flaws. (Paper "Forging of High Temperature Alloys" was presented at SAE Annual Meeting, Detroit, Jan. 10, 1949. This paper is available from SAE Special Publications Department. Price: 25¢ to SAE members, 50¢ to nonmembers.)

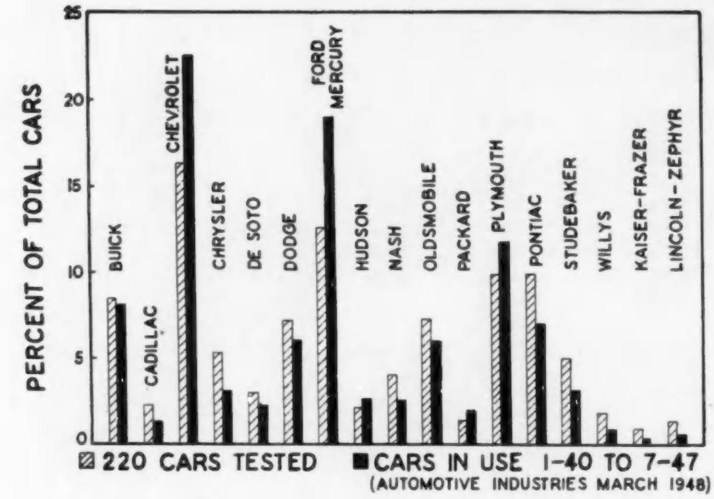


Fig. 1—Distribution of cars tested, by model, in Shell's 1947-1948 octane requirement survey

Propose New Method For Rating Gasoline

Based on paper by

F. R. Watson
F. H. Caudel
and J. D. Heldman

Shell Oil Co., Inc.

A practical method of evaluating road octane number of a gasoline has been developed which accurately reflects the customer's reaction to a fuel's antiknock characteristics. Reason: the fuels are tested where they are actually used—in the customers' cars.

The proposed road octane number satisfies the criteria of practical antiknock performance evaluation. Here is how:

1. It reflects the antiknock performance of fuels in the "population" of cars concerned.
2. It is convenient to use since it is a one-number expression.
3. It is built around a stable standard of antiknock quality—the octane scale.
4. It can be determined either directly by road testing, or indirectly from correlation established with the laboratory Research Method (F-1) and Motor Method (F-2) octane numbers.

This gasoline road octane number is defined as the percent iso-octane in normal heptane which will give knock-free performance in exactly the same percentage of cars in general as does the gasoline.

These are the four steps involved in determining this road octane number directly from road tests:

a. Select sample cars for the test. The number, age group, proportion of various makes, and geographical location of cars making up the sample will vary with the interests of the investigation. The car population is and should be flexible; the road octane number must continue to reflect relative antiknock quantity as the cars on the road undergo change.

b. Apply some method of determining the octane requirement of cars. The general method, CRC E-1, with minor modifications, has been found satisfactory.

c. Test the series of primary reference fuels (blends of iso-octane in normal heptane) in increments of five

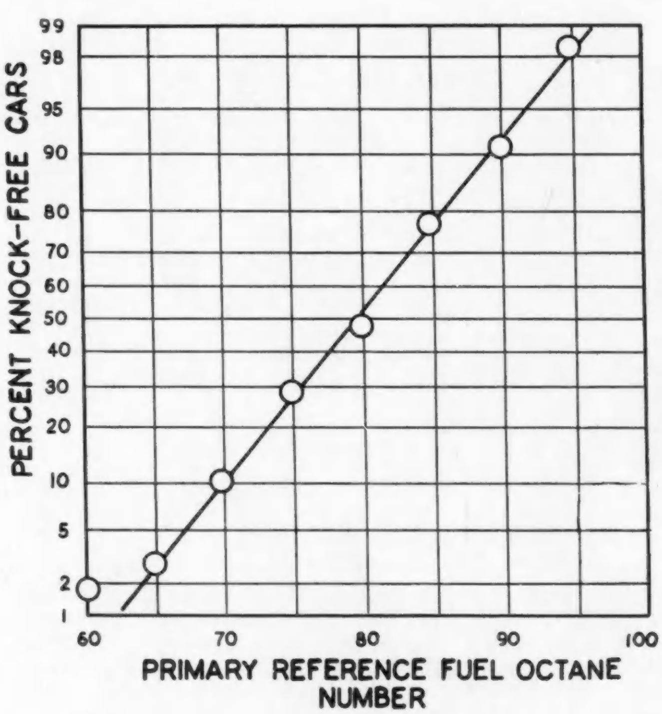


Fig. 2—Plot of test data on arithmetic probability paper showing the percent of knock-free cars for the various primary reference fuels, by octane number

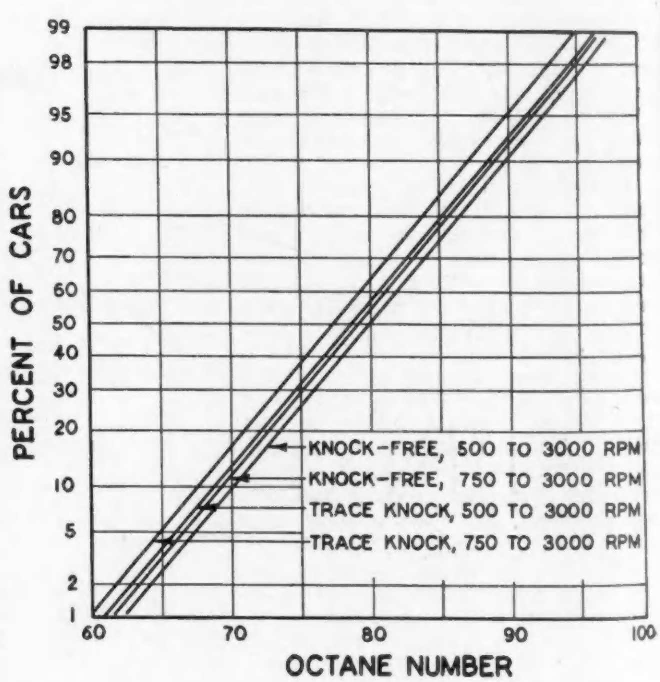


Fig. 3—This chart shows how octane requirements of cars vary with knock and speed-range criteria

octane numbers in each car along with the gasoline being rated. Cars are tested in the "as-received" condition. In each car this simple determination must be made for each reference fuel and gasoline: Is it knock-free or does it knock?

d. Calculate the road octane number for the gasoline after completing the tests. First, compute the percent of knock-free cars for each reference fuel and plot this as percent of knock-free cars versus octane number, using arthmetic probability paper for convenience. This should give a straight-line plot if a random sample has been chosen.

Knowing the percent knock-free for the gasoline, refer to the reference fuel plot to find the reference fuel octane number equivalent to that percent knock-free cars. The value obtained is the road octane number—the octane number of the primary reference fuel which gives knock-free performance in exactly the same percent of cars in general as does the gasoline.

Theory in Practice

This method of determining road octane numbers was used in a recent West Coast octane requirement survey, involving 220 cars, mostly 1940 to 1947 models.

Five commercial gasolines of about 5.5 to 8.5 sensitivity were tested in 148 of the cars, two commercial gasolines of 12.5 sensitivity in 34 cars, and the primary reference fuels in all 220 cars. Fig. 1 shows the distribution of cars tested by make.

Fig. 2 shows the percent knock-free cars various octane number for all the cars over a 500 to 3000-rpm speed range (about 10 to 60 mph). Note that using arithmetical probability paper produces a straight line, indicating that this peak octane phenomenon follows a normal distribution curve. These data show that 99% of the cars tested are knock-free with 97 octane reference fuel and 90% with 89 octane reference fuel.

Criteria Govern Results

Actual percentage of cars knock-free at the various octane levels depends on the knock and speed range criteria used. The criteria used in this investigation—no knock over the 500 to 3000-rpm speed range—are believed to be both conservative and precise. Less severe criteria, such as trace knock and/or restricted speed ranges, would naturally increase the percent cars "satisfied" at any one octane level, as shown in Fig. 3.

To get the road octane number indirectly from the laboratory F-1 and F-2 engine ratings, a practical correlation must be found. A simple and direct way of doing this is by using empirical equations.

Suppose we consider this relation-

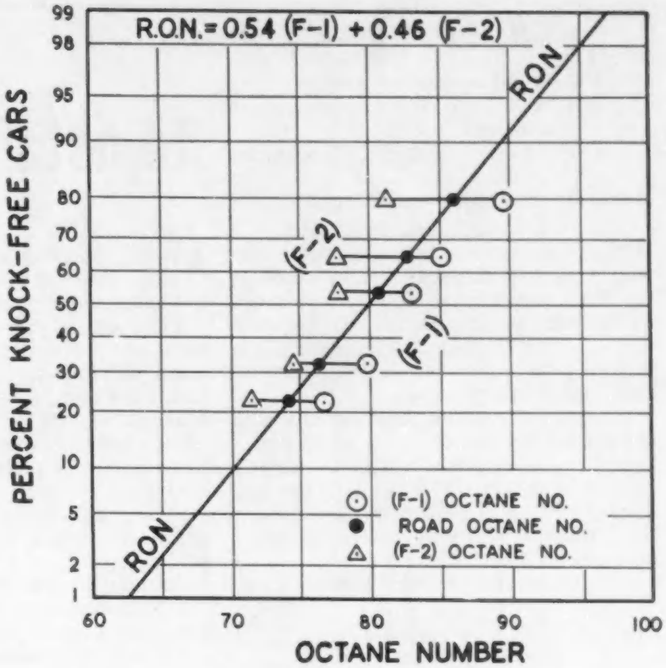
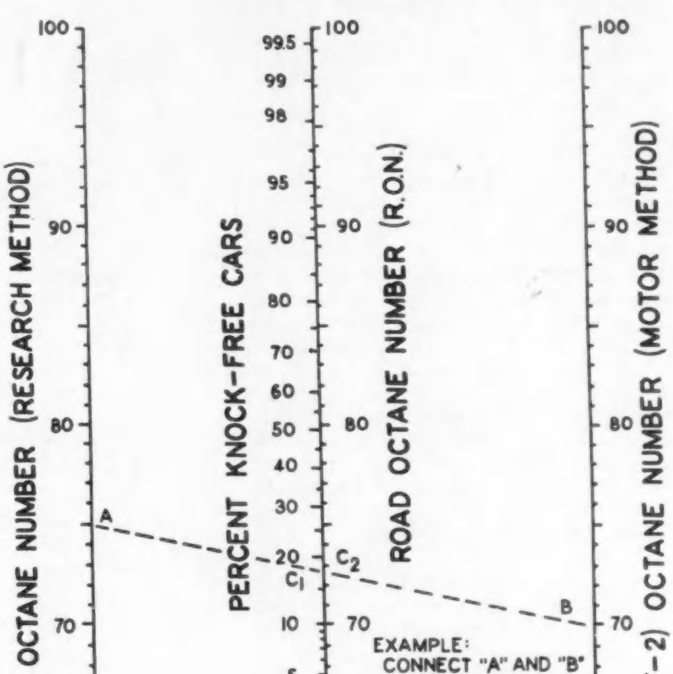


Fig. 4 (above)—Percent knock-free cars for five test gasolines in 148 cars

Fig. 5 (below)—Nomograph for predicting road octane number and percent knock-free cars



ship:

$$R.O.N. = A(F\text{-}1) + B(F\text{-}2) \quad (1)$$

where:

R.O.N. = road octane number,
F-1 = Research Method laboratory rating,
F-2 = Motor Method laboratory rating,
A and B = empirical coefficients.

This assumes that F-1 and F-2 laboratory numbers taken together correlate with the road octane number, and that the relative influence of those antiknock properties of a gasoline shown be F-1 and F-2 numbers are reflected by values of coefficients "A" and "B" respectively.

Any such general relationship also should fit the specific case of the primary reference fuels, which by definition have R.O.N. = F-1 = F-2. This can be true only if A + B = 1. This simplies the problem, since equation (1) may now be rewritten:

$$R.O.N. = A(F\text{-}1) + (1 - A)(F\text{-}2) \quad (2)$$

Assuming the relationship shown by equation (2), it is relatively simple to calculate the value of "A" for any given set of data by the method of least squares.

Using Fig. 2 as a basis, the percent knock-free cars was calculated for each test gasoline and a road octane number assigned, as shown graphically in Fig. 4. The least squares solution in this case yields the following correlation equation:

$$R.O.N. = 0.54(F\text{-}1) + 0.46(F\text{-}2) \quad (3)$$

Relationship between road octane number and laboratory ratings also can be expressed in terms of fuel sensitivity "S" as follows:

$$R.O.N. = (F\text{-}2) + 0.54(S) \quad (4)$$

$$R.O.N. = (F\text{-}1) - 0.46(S) \quad (5)$$

From these equations it is apparent that in comparing gasolines of the same F-2 rating, the one with the highest sensitivity will give the greatest overall percentage of knock-free cars. With gasolines of the same F-1 rating, the one with lowest sensitivity will give the greatest percentage of knock-free cars. The nomograph in Fig. 5 relates the road octane number and the percent of knock-free cars with the laboratory F-1 and F-2 octane numbers.

Certain cars are generally critical to F-1 quality and relatively unaffected by the F-2 quality of a gasoline; with others, the converse is true.

By clearly revealing these phenomena, the engine designer may be assisted in tailoring his engine antiknock requirements to gasolines, and the petroleum refiner may be helped in tailoring his motor gasolines to the needs of his customers' cars. (Paper "A Proposed Road Octane Number," was presented at SAE National Fuels & Lubricants Meeting, Tulsa, Nov. 5, 1949. This paper is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

Adjustable Coil Springs Enhance

Based on paper by
WILLIAM A. CLARK
L. A. Young Spring & Wire Corp.

A new adjustable coil-spring construction for car cushions offers new levels in comfort at lower cost.

Fully adjustable to suit the passenger's or/and driver's physical proportions, the new design offers soft initial feel, high damping action, low rebound tendencies, and is noiseless. It banishes objectionable "hammock" action by using resilient helical coil ties. This new construction costs less than the six-row, coil-spring type design because it weighs less and has fewer parts.

Foundation for this new innovation is a grid-like wire construction, Fig. 1. The horizontal and vertical border wires are spotwelded together and to a border member forming a sturdy subassembly. Novel arrangement of these wires minimizes the number of operations for attaching the pocketed coil springs.

The coil springs, of which there are four rows, are knotted and pocketed in burlap and mounted to the bottom frame with hog rings to prevent dislocation. (See Fig. 2.) Eliminating two rows of pocketed coils reduces weight and saves more than two yards of expensive burlap; this reduces costs.

The top border frame is made of two flat-wire rims, one inside the other. They are securely held together by strut-like spacing members, with enough pressure applied to indent the rims and prevent slippage. This top border frame holds contour lines by not yielding horizontally, yet is vertically resilient.

The newly-designed diagonal brace members, with an offset and spiral loop

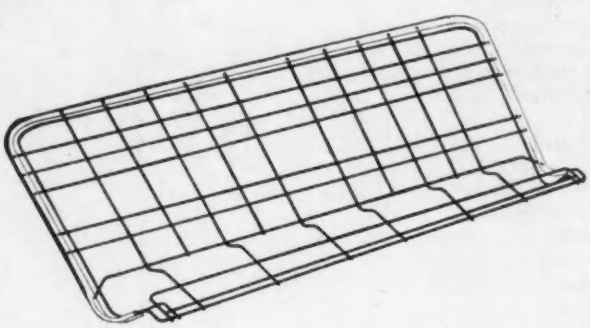


Fig. 1—A spotwelded wire grid structure forms the foundation of the adjustable coil-spring seat cushion

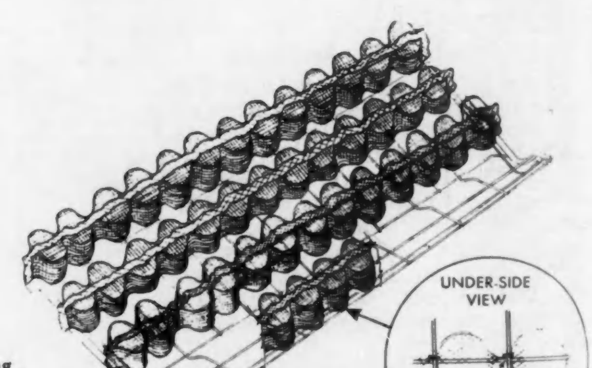


Fig. 2—Tying each coil spring to the bottom frame with hog rings keeps them in place

Seating Comfort

(or pigtail) at one end and a hook at the other, in opposite planes, provide balanced sidewise stability. This is shown in Fig. 3. The braces are positioned between the coil springs to allow noiseless free action. In most six-row cushion applications, the braces project through the coils, which restricts their action and causes a crunching noise.

The resilent heat-treated coil ties are secured to the top of the coil springs, as in Fig. 4, and have three definite functions: (1) they position the coil springs, (2) give a soft initial feel, and (3) give the occupant added resilient support after static depression by his body weight.

A pad protector of oil-tempered wires, woven through burlap applied to the top of the unit prevents the padding from working down between the coils. This functional structure, as shown in Fig. 4, now is ready for the padding and trim cloth.

Coil-spring front cushions can be designed with ample toe room for rear seat passengers. Many people are allergic to touch or pressure to the sitting part of their person. Toe-room provision of this cushion eliminates any possibility of the toe or foot of rear seat riders contacting in any way the sitting portion of the front seat occupants.

The manufacturer who has equipped his car with adjustable seating units can offer a choice of comfort through his nationwide service stations.

For example, consider a man and wife, he of average height but heavy;

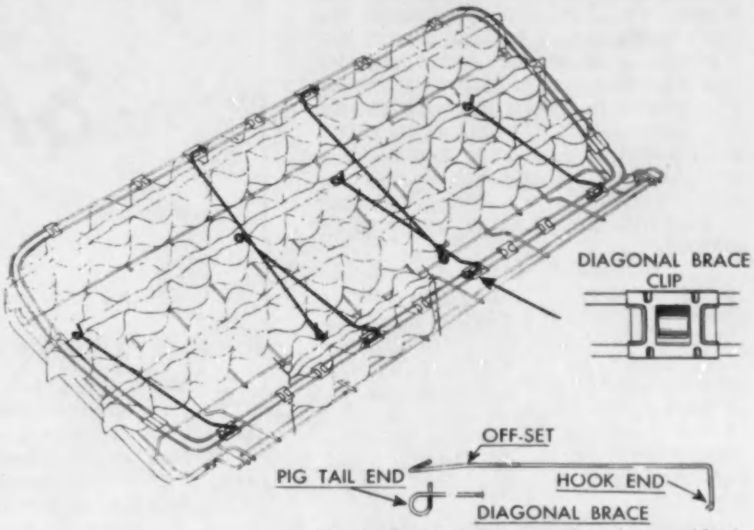


Fig. 3—Diagonal brace members impart sidewise stability to the adjustable coil-spring seat cushion. They are positioned to eliminate the noise experienced with the luxury-type construction

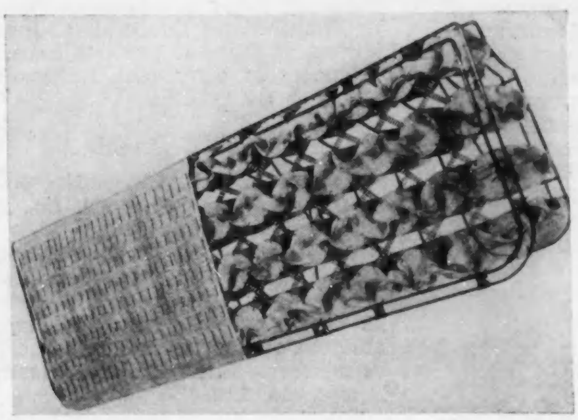


Fig. 4—Resilient heat-treated coil ties attached to the coil springs provide an added measure of comfort

Fig. 5—The coil-spring cushion can be adjusted to give added support by inserting four to six pocketed coils of prescribed firmness

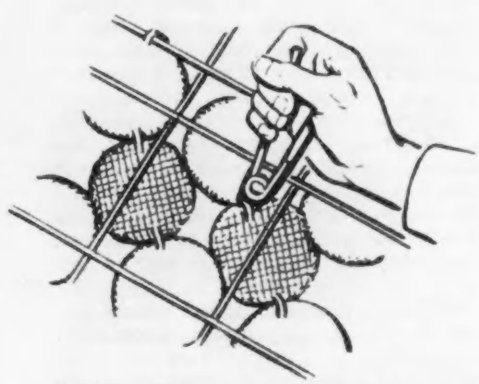


Fig. 6—In adjusting the coil-spring seat cushion, it is advisable to hog ring each insert coil in position

she below average on both counts. As in the case of most passengers, the woman would enjoy the luxury of the unadjusted seat. As the driver, her husband may prefer added support and increased vision. After the proper adjustment has been made to satisfy his requirements, the wife (when she drives) will be pleased to find her eye level has been raised an inch or more, and yet because the "firmed-up" seat is buoyant, she remains comfortably seated.

The cushion is easily custom-adjusted in about 3 min. The service man simply lifts the cushion assembly from the seat frame and places it, bottom side up, on a bench. (The upholstery is in no way disturbed.) Each of four to six pocketed coil springs, of the prescribed degree of firmness taken from the dealer kit, is compressed and inserted into the cushion, as in Fig. 5.

In most designs it usually is desirable to hog ring each insert coil in position, Fig. 6, even though the extra coils nest perfectly and are supported by the wires of the foundation frame. Now the cushion, having received custom treatment, is returned to the seat frame ready for luxurious enjoyment. (Paper "Custom Riding Comfort with Coil Springs," was presented at SAE National Passenger Car Body, and Production Meeting, Detroit, March 8, 1949. This paper is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

Sea and Land Rescue Advances Under Way

Excerpts from paper by

LT.-COM. A. W. WUERKER

U. S. Coast Guard

SHORTCOMINGS of search and rescue of air and sea travelers will be minimized in time through national and international coordination, and through development of adequate procedures, equipment, and facilities.

With search and rescue ranging from Arctic to desert operations, obviously special problems arise. Both the icecap and the open sea demand special equipment and techniques. It would be foolhardy and fatalistic not to anticipate repititions as well as new and difficult situations.

Effective search depends largely upon the communications equipment in the hands of the potential survivor. If he can make his position known to the world, he will be found eventually. For this reason proper radio direction-finding stations guarding appropriate

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Officers 1949–1950

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Chairman: **Edgar I. Billings,** sales engineer, Socony-Vacuum Oil Co., Inc.

Vice-chairman: **Lester Anthony;** secretary-treasurer: **A. Frank Geiler,** president, general manager, Schenectady Railway Co.

Salt Lake Group

Chairman: **Frank G. Backman,** manager, Midwest Service & Supply Co.

Vice-chairman: **Lyle G. Garnas,** salesman, International Harvester Co.; secretary-treasurer: **Stanley W. Stephens,** machine shop foreman, Koepsel & Love.

Williamsport Group

Chairman: **Robert B. Ingram,** experimental engineer, Lycoming Division, Anco Mfg. Corp.

Vice-chairman: **James C. McRoberts,** assistant project engineer, Lycoming Division, Anco Mfg. Corp.; treasurer: **John W. Hospers,** analytical engineer, Lycoming Division, Anco Mfg. Corp.; secretary: **Carroll S. Townsend,** chief engineer, General Armature & Mfg. Co.

DAVIES

ROSEN

J. M. DAVIES, associate director of research at Caterpillar Tractor Co., has been named director of research, succeeding **C. G. A. ROSEN** who is showing progress in recovery from a recent illness. Davies has been with the company since 1925 when he joined the firm at San Leandro, Calif., as a laboratory engineer. He later became research engineer, assistant chief engineer, and in 1942 became assistant director of research. His appointment as associate director of research came in 1945. Rosen, in an advisory capacity, will devote his time to the further development of Diesel engine design and performance and will aid in the advancement of the company's technical program. He joined Caterpillar in 1928 as a consultant and was their first director of research; the department being organized in 1942

About

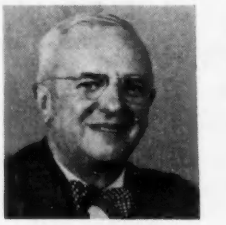
EDWARD J. ALSHUT, JR., is now senior draftsman at the Goodyear Aircraft Corp., Akron, Ohio.

LEROY A. HOWARD, now instrumentation engineer at Sverdrup & Parcel, Inc., St. Louis, Mo., had been connected with the Aircraft Division of Packard Motor Car Co., Toledo, Ohio.

OTTO AUGUST SCHOLZ recently became vice-president of the American Disc Brake Co., Inc., Berkeley, Calif.

FREDERICK E. NEEF, JR., now sales engineer of industrial sales for the Gilbert & Barker Mfg. Co. in West Springfield, Mass., had been equipment development engineer in the Marketing Department of the Esso Standard Oil Co., New York City.

HOWARD C. WORDEN is project engineer at the Fruehauf Trailer Co. in Detroit. His previous position with this company was assistant resident engineer at Avon Lake, Ohio.



CHARLES H. STANARD, since 1944 productive gear engineer of Buick Motor Division, General Motors Corp., Flint, and chairman of the SAE Subcommittee on Involute Splines and Serrations, under the SAE Parts and Fittings Technical Committee, has retired and will continue to live at Flint, Mich. He is secretary of Technical Committee 13 on Involute Splines and Serrations, the ASA Sectional Committee B5, Small Tools and Machine Tool Elements.

TOM J. COLLINS has resigned as western regional automotive manager of Buda Co. and has formed the Tom Collins Mfg. Co., 793 East 17 St., Los Angeles 21, Calif. The company will manufacture and distribute diesel engine parts. Collins is vice-chairman of the SAE Southern California Section, having served as membership chairman and vice-chairman for Diesel Engines.

PAUL B. HARTMAN, assistant director of research at Willys-Overland Motors, has been elevated to the post of director of research. He joined the company in 1943 as an experimental engineer in powerplants. He aided in improvements in the military Jeep and helped develop the civilian Jeep and other vehicles in the Willys-Overland line.

SAMUEL F. ROLPH will continue as general manager of the Norton Door Closer Co. Division in addition to his new duties as general manager of Sager & Barrows Lock Works Division, both of Yale & Towne Mfg. Co., Chicago.

J.G. MORROW, metallurgical engineer, The Steel Co. of Canada, Hamilton, Ont., was installed president of the American Society for Testing Materials at its 52nd Annual Meeting at Atlantic City, N. J. **VAN M. DARSEY**, president of the Parker Rust Proof Co. was one of the winners of the Sam Tour Award for his paper "Apparatus and Factors in Salt Fog Testing."

JOHN B. BLACK, until recently research engineer with Glenn L. Martin Co., has established the John Black Engine Co., Inc., Garrett, Pa., and is president of the board and chief engineer. He had been in charge of engine design and installation of the Northrop Flying Wing, and had had important engineering assignments at United Aircraft Corp., Consolidated, and Douglas. His company is developing and manufacturing aircraft components.

EARLE BUCKINGHAM, professor of mechanical engineering, Massachusetts Institute of Technology, is the author of "Analytical Mechanics of Gears," recently released by the McGraw-Hill Book Co., Inc. It is a source book rather than a text on fundamental relationships in the design of all gear types. Uniform transmission of motion and power by gears is covered in its entirety. The book deals with gear action, tooth forms, nature of friction, conditions influencing intensity of dynamic loads, strength of gear teeth, and resistance to wear of various combinations of materials.

MUNSON HENRY TIX is industrial engineer at the U. S. Ordnance—Reo River Arsenal, Texarkana, Tex.

Members

Recent graduates from Purdue University included **RICHARD JOHN WHEATON**, detail engineer, J35 Design Group, Allison Division of General Motors Corp., Indianapolis, Ind.; **DELOS M. SUTTER**, experimental test mechanic, Electro-Motive Division, General Motors Corp., LaGrange, Ill.; **HERBERT JOHN ANDERSEN**, junior test engineer, International Harvester Co., Melrose Park, Ill.; **MARVIN E. OLDS**, a member of the Engineering Test Department, General Motors Proving Ground, Milford, Mich.; and **ROBERT C. SWANSON**, now affiliated with The Peoples Gas Light & Coke Co. in Chicago.

STANWOOD W. SPARROW, Studebaker's vice-president of engineering and 1949 SAE President, has received an honorary degree of Doctor of Engineering from Worcester Polytechnic Institute, his alma mater. He has also been elected to the Board of Trustees of WPI. The citation accompanying the degree reads in part: "His eminence in science and engineering is not exceeded by his rare competence in bringing the creations of the laboratory to factory production. . . ."

ALFRED P. SLOAN, JR., chairman of the board, General Motors Corp., has contributed $1,000,000 to Massachusetts Institute of Technology for construction of a Metal Processing Laboratory building. Sloan was graduated from MIT with a Bachelor of Science degree in 1895.

K. T. KELLER, president, Chrysler Corp., will be guest of honor at the Automobile Old Timers' celebration of its 10th anniversary at a dinner at the Hotel Astor in New York City on Oct. 18.

GEORGE A. BLEYLE is now lubrication engineer in the Railway Sales Department of the Sinclair Refining Co., New York City. He was formerly with the Wright Aeronautical Corp. He was chairman of the SAE Committee on Cold Starting Requirements for Aircraft Engines.

EUGENE PHILLIP WELCHER is now field service representative on transmissions and torque converters for the Allison Division of General Motors Corp., Indianapolis, Ind.

GERALD A. PETERSON recently became senior cost analyst at the Ford Motor Co. in Dearborn, Mich. His previous connection was at the Boeing Airplane Co. in Seattle, Wash.

BURNHAM ADAMS, manager of the California Division of Lear, Inc., Grand Rapids, Mich., since May, 1946, was recently elected vice-president of the corporation. Prior to this election, he had supervised the sales and manufacturing activities of the California Division in Los Angeles. Before joining Lear, Adams was at Wright Aeronautical Corp. for 22 years, having been sales manager and later West Coast manager.

EDMUND B. NEIL, consulting engineer in Columbus, Ohio, has been appointed midwest associate for **FRANCIS W. DAVIS**, consulting engineer in Waltham, Mass. Davis is engaged in the development of power-assisted steering and other hydraulic devices for motor vehicles. Previously Neil and Davis were associated at the Pierce-Arrow Motor Car Co., Buffalo, N. Y., since after the end of World War I.

HUBERT C. SMITH, Great Lakes Steel Corp. chief metallurgist, has been appointed assistant vice-president in charge of metallurgical control, a new position created in recognition of increased responsibilities for production quality which have accompanied the Ecorse, Mich., plant's expansion program. Smith is a member of the SAE Engineering Materials Activity Meetings Committee.

T. EARL WAGAR, Studebaker electrical engineer, shows to **J. H. BOLLES**, Delco-Remy chief engineer, the watch which he received recently from Studebaker Corp., in recognition of 40 years of service. Wagar is a member of the SAE Electrical Equipment Committee of which Bolles is chairman, and is himself chairman of the Subcommittee on Motors and Generators.

GORDON D. BROWN, formerly vice-president of the Bankers Trust Co. in New York City, has established a financial and management consultant firm under the name of Gordon D. Brown & Associates. The company is operating on a national basis from offices in Los Angeles, but shortly expects to open offices in New York. They will specialize in financial planning and management controls.

ELMER W. KRUEGER has been appointed operations manager of the Cleveland Pneumatic Tool Co. He will be in charge of the departments of sales administration, engineering, purchasing, inspection, production, mechanical and maintenance operations, and industrial relations. Krueger joined Pneumatic in 1922.

H. FLETCHER BROWN, factory manager of the Airplane Division of Curtiss-Wright Corp. in Columbus, Ohio, has been appointed general manager of the division. Brown joined the organization last year and was previously vice-president in charge of production manufacturing for Boeing Airplane Co. During the war he was works manager of the Boeing plant at Wichita, Kans.

JAMES L. DOOLEY has been named chief engineer of Rhodes Lewis Co., Los Angeles, designers and manufacturers of aircraft and automotive accessories for the Armed Services. He was formerly retained as a consulting engineer by Rhodes Lewis Co. and North American Aviation's Aerophysics Laboratory, where he specialized in the mechanical and control problems of propulsion equipment in large rocket missiles.

HERBERT R. JAFFE, formerly with the General Electric Co., Schenectady, N. Y., is now automobile designer with the General Motors Styling Section.

CHARLES R. McCLELLAN has become head fuel and lubricant engineer with the Standard Oil Co. of California in Honolulu, T. H. He had been fuel and lubricant engineer with the company in Long Beach, Calif.

HAROLD C. WATERHOUSE is salesman with Mack International Motor Truck Corp., Atlanta, Ga.

B. MORRIS HENDERSON, recently graduated from the University of Illinois, is a junior research engineer in the Research Laboratory of Shell Oil Co., Wood River, Ill.

GEORGE H. LANCASTER is now associated with the Chicago Transit Authority on problems connected with automotive engineering aspects of transportation equipment. Formerly he was connected with the Magnolia Petroleum Co. in Beaumont, Tex.

CHARLES W. TUCKER is now marine fuel and lubricant representative for Esso Export Corp. in Norfolk, Va. Prior to this post he was an industrial salesman for the Esso Standard Oil Co. in Richmond, Va.

JOHN W. ANDERSON, consulting engineer, is the author of "Diesel Engines," the second edition of which recently was published by the McGraw-Hill Book Co., Inc. This book covers automotive, marine, railway, industrial, and stationary diesel powerplants. It describes them, tells how to select and operate them in the light of current practice. Several chapters are devoted to trends in component design, including combustion chambers, fuel injection, cooling, fuel, and lubricating systems. (In the section "About SAE Members," p. 78 of the June, 1949, SAE Journal, John W. Anderson, author of the book "Diesel Engines," was erroneously listed as John W. Johnson.)

JACK R. STITT is now technical representative for the Oronite Chemical Co. in New York City. He had been research engineer, California Research Corp., Richmond, Calif.

RONALD D. SPEED recently became assistant works manager for Robb Motors, Ltd., Cape Town, South Africa.

HERMAN HORTON BEMENT recently joined the Grumman Aircraft Engineering Corp., Bethpage, L. I., N. Y., as flight test powerplant engineer.

BRUCE O. TODD is now sales and engineering representative for the Metal Products Division of Ryan Aeronautical Co., San Diego, Calif.

HENRY FORD II, dinner speaker at the Yale Alumni Fund Association recently said: ". . . the physical scientists can get us into trouble they can't get us out of. . . . It may be that the biggest problems of our times will not be solved by scientists but by an increased recognition of the importance of moral and spiritual leadership and character."

RALPH S. DAMON, president of Transcontinental & Western Air, Inc., has won the American national trophy of the International League of Aviators for outstanding contributions to aviation over 31 years. The trophy was established in 1925 by the late Clifford B. Harmon, pioneer balloonist, and other well known aviators.

SUTHERLAND LAUCK

JOHN A. LAUCK has been named vice-president of Pesco Products Division of Borg-Warner Corp. having been chief engineer of Pesco's pump division for the last four years. He has been associated with Pesco since 1937 as test engineer, chief test engineer, and assistant chief engineer. **D.A. SUTHERLAND,** formerly Pesco's Eastern sales manager for industrial products, has been appointed industrial relations manager. He has served as field engineer in charge of sales and sales promotion for the B-W Supercharger Division of Borg-Warner, and earlier was power engineer and sales engineer for Fairbanks Morse & Co.

JOHN D. TEBBEN has announced the forming of a company to do consulting work in engineering sales and industrial relations, with headquarters at 20869 Mound Road in Detroit. Tebben has been active in the metallurgical field for many years, and is nationally known in the electrical, aviation and resistance welding industries. He was formerly general sales manager for P. R. Mallory Co. and vice-president of S.M.S. Corp. During the war he was a colonel in the Air Corps, stationed at Wright Field, Dayton, Ohio, as chief of the Aircraft Production Branch.

KENTON ARNEAL STOCKBARGER is now experimental technician in fuel cells for aircraft at the Firestone Tire & Rubber Co., Los Angeles.

ADOLF T. R. SAUL recently became a mechanical engineer at the U. S. Bureau of Reclamation, Casper, Wyo.

JOHN P. BERTRAM is now associated with the Fleet Sales Division of the J. B. E. Olson Corp. in Brooklyn, N. Y., distributors of truck bodies designed by Grumman.

LEROY A. DIFFORD has become an engineer at the Textile Equipment Corp., Greenville, S. C. He was previously a development engineer at the Weatherhead Co., Cleveland, Ohio.

DALE L. DRAKE is now draftsman with the Illinois Power Co., Decatur, Ill.

HARRY R. REYNOLDS has been appointed consulting engineer at the Fafnir Bearing Co., New Britain, Conn. His previous position was chief engineer. He is one of the pioneers in the American ball bearing industry, having been with Fafnir since the company's formation in 1911.

K. A. KREKORIAN has become flight test analyst at North American Aviation, Inc., Inglewood, Calif.

DONALD F. MORGAN is now stress analyst at the Schweizer Aircraft Corp. in Elmira, N. Y.

HERBERT B. SMITH recently became a partner in the Brake Service Co. in Winston-Salem, N. C. He has resigned his position as regional manager with Bendix Westinghouse Automotive Air Brake Co.

ROBERT L. NELSON is now template maker with the Chance Vought Aircraft Co. in Grand Prairie, Tex.

WILLIAM A. M. BURDEN, former assistant secretary of Commerce, and aviation consultant with Smith, Barney & Co., New York City, has been elected a trustee of Central Hanover Bank & Trust Co., same city. Burden is president of the Institute of the Aeronautical Sciences, Inc.

SIR ROY FEDDEN has been appointed director of research of Leyland Motors Ltd., London, England. He will be responsible for building up the company's new long term research program. During World War II he was special technical advisor to Sir Stafford Cripps then minister of aircraft production. Twice president of the Royal Aeronautical Society, Fedden won the SAE Manly Memorial Medal in 1933, the Guggenheim Trophy in 1938, and a number of high British and European awards for his contributions. He received the degree of Doctor of Science from Bristol University, and was elevated to knight bachelor in 1942 by the King of England.

Practical help for the designer of 2-stroke cycle diesel engines is contained in a recently published book by **PAUL H. SCHWEITZER.** In this book, "Scavenging of Two-Stroke Cycle Diesel Engines," the author treats only the subject of scavenging, as he considers skill in design of scavenging the key to successful 2-stroke cycle engines.

Readers interested only in practical applications can easily bypass the higher mathematics, which has been separated from the main text and placed in appendixes. Charts frequently replace complicated formulas. Many numerical examples illustrate the application of both formulas and charts. Trouble-shooting types of tests are also included for the test engineer. The book is published by Macmillan Co., New York City, for $7.25.

ROBERT H. SPAHR, JR., is a junior engineer at the AC Spark Plug Division of General Motors Corp., Flint, Mich.

GLEN A. SMITH has become product engineer with General Motors Overseas Operations in Detroit.

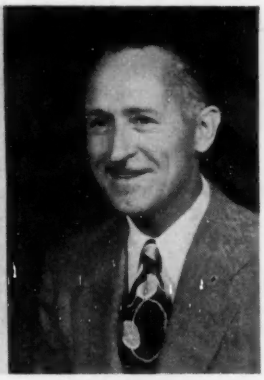
WAYNE B. GUTHRIE is now a sales engineer with the R. C. Hardcastle Co. in Oklahoma City, Okla. The company is a distributor of construction machinery, cranes, shovels, water and sewage equipment.

JOHN B. CLARK has become superintendent of Portland shops for Consolidated Freightways, Inc. in Portland, Oreg.

MARION F. SMITH is now a consulting engineer, located at 419 S. Santa Fe, Vista, Calif. He had been division engineer at ACF-Brill Motors Co., Philadelphia, Pa.

KEITH L. MIKESELL, left, and J. B. COLLETTE with a part of the equipment of the hydraulics laboratory they designed, built and described for their joint thesis for their bachelor of science degrees at California State Polytechnic College, San Luis Obispo. Mikesell will continue at the college to earn his graduate degree

MAUCK



WERNIG

P. J. MAUCK, former general director of the engineering division of Fisher Body Division, General Motors Corp., has been appointed executive assistant for engineering, general manager's staff. JAMES H. WERNIG, former assistant general director of the engineering division, was named to succeed Mauck as general director

RALPH B. WITTMAN, now associate engineer at Reaction Motors, Inc., Dover, N. J., had been stress analyst for Ranger Aircraft Corp. in Farmingdale, L. I., N. Y.

CALVIN J. BRESSLER recently became a design engineer at Cummins Engine Co., Columbus, Ind. He was previously group leader in the Drafting Department of Packard Motor Car Co., Aircraft Engine Division, Toledo, Ohio.

ERNEST L. KORB has become manager of the Technical Service Department, Marketing Division of the Pure Oil Co. in Chicago. He is a past field editor of the SAE Philadelphia Section.

VINCENT ELLIS has joined the Eastern Division of Lear, Inc., as sales engineering representative. Prior to this post, he was with Lord Mfg. Co. for nine years, as manager of the New York sales office and later the aircraft sales home office in Erie, Pa. Ellis will have headquarters in New York City and will handle sales activities in the Washington, D. C. area of Lear electro-mechanical devices for aircraft.

JACK L. O'CONNELL is an engineering draftsman at Lockheed Aircraft Corp. in Burbank, Calif.

ROBERT M. WARD has become sales manager of the Special Products Division, Thompson Products, Inc., Cleveland, Ohio. He was formerly Eastern representative for this company.

JOSEPH F. REGAN is research engineer at Pratt & Whitney Aircraft Division, United Aircraft Corp., East Hartford, Conn.

JOHN R. GRIFFIN, JR., recently became chief engineer of the Research Laboratory of the Sharples Corp., in Bridgeport, Pa.

JOHN G. RUSSELL is now a development engineer at the Parker Appliance Co., Cleveland, Ohio.

J. J. BIGELOW is now sales manager of the Automotive Division at the Sparks-Withington Co. in Jackson, Mich. Prior to this post, he was merchandiser in the Automobile Accessory Department at Goodyear Tire & Rubber Co., Akron, Ohio.

WILLIAM WADDELL is now with the American Locomotive Co., Schenectady Division at Schenectady, N. Y., as a design engineer.

W. N. GROVES has become a junior research engineer for the Shell Oil Co., Inc., Wood River, Ill.

LAWRENCE V. MELLO is a technical assistant for the Self-Locking Carton Co. in Palmer, Mass.

BURT M. ROWE recently became mechanical research engineer on rockets for the Bell Aircraft Co. in Buffalo, N. Y.

M. CONNER AHRENS is assistant agricultural engineer, United States Department of Agriculture, Bureau of Plant Industry, Soils and Agricultural Engineering, Washington State College, Pullman, Wash. He is doing refrigeration research in the State of Washington.

JOHN A. STALLINGS is a student engineer at the Buick Motor Division, Flint, Mich.

KENNETH N. WIERSEMA recently became junior engineer on powerplants at the Aluminum Co. of America, Pittsburgh, Pa.

WILLIAM J. GORMAN, JR., is now sales manager at ETC, Inc., Niles, Mich., manufacturers of electrical coils and punch press specialties for the automotive and radio industries. Previously he was connected with the Eclipse Machine Division of Bendix Aviation Corp., as sales engineer.

WILLIAM C. SCHMITZ has become junior industrial engineer at American Steel & Wire Co., Cleveland, Ohio.

WALTER A. HOOT has been appointed used car and truck manager of the Ford Motor Co., Southwest Region, Kansas City, Mo. He had been district representative of the company at Houston, Tex.

OBITUARIES

JOHN A. KURLANDER

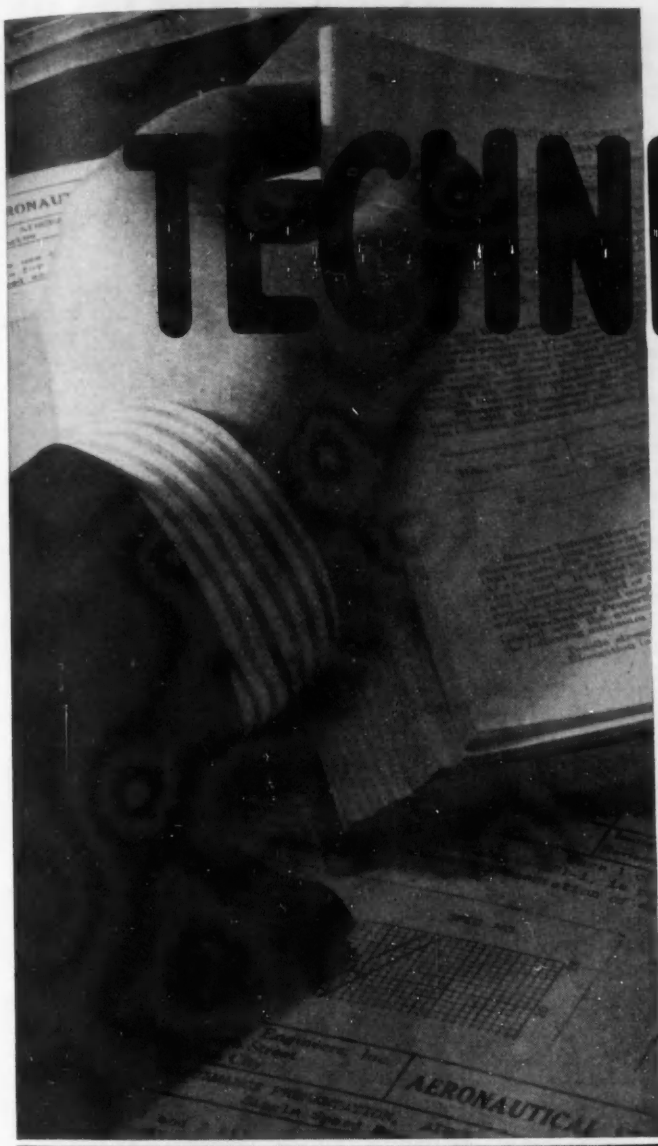
John A. Kurlander, Westinghouse Electric Corp., died of a heart attack at his Nutley, N. J. home on June 24. He was head of the projection, photographic, and miniature lamp section of Westinghouse's commercial engineering department and specialized in development of miniature lights for photographic purposes.

During World War II, he developed a gunsight lamp which eliminated the blind spot American airmen encountered in firing at enemy planes which were diving out of the sun.

He is also credited with many peacetime developments, including a device that produces either a spot of light or a flood of light from an ordinary hand flashlight; a blue bulb photoflash lamp emitting invisible, unobtrusive light; black light illumination for airplane instrument dials; and colored filter glass for automotive turn signals.

Born in Trenton, N. J., he graduated from Drexel Institute of Philadelphia and joined Edison Lamp works in 1920. Six years later, he became chief engineer of Brenkert Light Projection Co. in Detroit—and became associated with Westinghouse in Bloomfield in 1929.

He was active in professional societies, having served as secretary of the Society of Motion Picture Engineers for eight years and as a member of SAE's Lighting Committee for a number of years. He was also a member of the Illuminating Engineering Society and the American Optical Society.



TECHNICAL COMMITTEE PROGRESS

SAE TECHNICAL BOARD

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SAE Engine Group Plans Standards on Broad Front

THE SAE Engine Committee's program is moving ahead on such projects as standards for governor spacing dimensions, carburetor air horns and flanges, hydraulic pump mountings, fan mounting sizes, and engine nomenclature.

Subcommittees on these jobs have submitted progress reports to Chairman H. S. White, Ford Motor Co., outlining general problems and recommendations on specifics to be tackled.

The group on throttle-type governors is considering standardization of space requirements and flange mountings for such governors now used on cars and trucks. Such standard would give engine and body designers the space limitations for proper engine governing without hood or air cleaner interference.

This subcommittee, headed by W. Malecki, King Seeley Corp., suggests a chart that will carry all physical dimensions of a composite or universal governor. From this the designer could get all the necessary installation information to complete his layout and provide for any of the presently available throttle-type governors.

Also recommended by the subcommittee is an extension of its scope to include standardization of nomenclature, performance specifications, provision for spark advance, provision for power jet transfer or by-pass holes, and sealing or tamperproofing.

A. H. Winkler, Eclipse Machine Division, Bendix Aviation Corp., chairman of the subcommittee on carburetor air horns and flanges, reports progress in his area. His group is collecting and screening

H. S. White

Chairman, SAE Engine Committee

data to develop uniform air entrance and manifold flange dimensions.

It also sees as practicable at this time a standardization program covering these items:

1. Separate standards for up and down-draft carburetors.

2. Separate standards for heavy-duty and normal-duty installations.

3. Air cleaner attachment flanges of the telescope type, "V" type, cap screw type, and modified telescope type for tube extensions.

4. Revision of carburetor manifold attachment flanges for both single-barrel and double-barrel carburetors.

5. Uniform identification of carburetor bore and flange.

A third project, standardization of hydraulic pump mountings, has been stimulated by the growing use of hydraulic power take-offs. Investigation of this subject by J. F. Greathouse, International-Plainfield Motor Co., reveals that the diversity of pump (or motor) types in use together with specialized automotive equipment using hydraulic pumps may limit the value of an arbitrary standard.

But Greathouse does point out that if a standard is necessary to guide those designing new equipment or modifying present designs, it is possible to set up a series for flange-mounted pumps covering a range of sizes in square, round, and triangular shapes. He feels that bolt holes, bolt circles, and pilot proportions can be made common to all shapes according to flange sizes.

It seems that at least two types of shaft end designs (in various sizes) will be necessary—one having a Woodruff key slot with optional threaded shaft end for a securing nut or a set-screw flat, and the other using SAE involute splines in the fractional inch sizes within ball bearing hole ranges, with snap ring grooves for securing the drive components.

The wide variety of fan mounting flange dimensions presently in use indicates a need for standardization, says Beecher Cary, Hayes Industries, Inc., chairman of the subcommittee on this subject. He reveals that minor variations in bolt circle diameter, number of bolts, and center hole size make for this variety. One manufacturer alone is producing over 60 different fan flange mountings. It is felt that many fewer mountings could satisfactorily replace all of these.

In addition to this work under way, Chairman White is organizing a Subcommittee to develop a standard engine nomenclature. Another will be created to standardize water pump seals. A study of oil filter mountings also is in progress.

Technishorts . . . .

SOUND METER: A new three-band sound meter for measuring motor vehicle noise is being studied by the SAE Automotive Traffic Noise Subcommittee. Built by E. J. Abbott, Physicists Research Co., in accordance with specifications developed by the Subcommittee, this instrument has three simultaneous-reading meters that are said to give satisfactory analysis of sound and correlations with ear observations. This instrument was tested on a highway near Monroe, Mich., reports Subcommittee Chairman F. B. Lautzenhiser, and a final report on the results is being prepared.

NEW NAME: The SAE Motorcoach and Motor Truck Committee has been renamed the SAE Truck and Bus Technical Committee. This change makes the Committee's name consistent with the SAE Commercial Vehicle Nomenclature Standard. W. P. Michell, Spicer Mfg. Division, of Dana Corp., heads up the Committee.

BODY NOMENCLATURE: The SAE Body Engineering Committee has appointed a subcommittee under W. Robertson, Budd Co., to develop a nomenclature for body and sheet metal parts.

LOW-TEMPERATURE BEHAVIOR: An extensive report on low-temperature properties of ferrous materials is being prepared by a division of the SAE Iron & Steel Technical Committee. Embrittlement of some irons and steels in subzero applications calls for careful attention to both design and material selection for military, marine, and civilian equipment. Information for designers and metallurgists facing such problems will be included in the report. The group doing this job is headed by C. H. Lorig, Battelle Memorial Institute.

Seek Measuring Tool For Vehicle Exhaust

IN its search for a simple yet reproducible method of measuring motor vehicle exhaust smoke, the SAE Subcommittee on Measurement of Automotive Exhaust Smoke has studied several techniques and found most have disadvantages.

These methods were evaluated at a recent meeting of the Subcommittee by Chairman J. C. Miller, Cummins Engine Co. He discussed the Ringelmann Chart, the CRC Photo Volt smokemeter, a polaroid device, a suction pump, photographs, and a precipitation instrument.

In the Ringelmann Chart method, the smoke is matched against a chart with various shades of gray and rated by the Ringelmann Chart tone of gray it most resembles. Trouble with this method, Chairman Miller pointed out, is that sky background condition and intensity of light affect these visual observations.

For example, smoke which may look almost invisible on a bright day against a brilliant blue sky will be quite apparent against white clouds, and even more so on an overcast day with a gray background. This makes it necessary to temper smoke appraisal—used in combustion studies as a measure of completeness of combustion—with judgment and mental correction factors.

Objections to the CRC Photo Volt smokemeter for this work are that it is too cumbersome, requires too much attention, and is too dependent on its placement in the exhaust line. It is considered purely an academic tool. The CRC instrument is essentially a tube with a light source at one end and at the other a photocell, connected to a microammeter graduated in percent smoke. Introducting the exhaust sample into the tube cuts down light intensity and the meter deflects in proportion to smoke density.

A small device made with two sheets of polaroid proved interesting for comparing succeeding readings or for evaluating two stacks side-by-side. But for making readings from one day to another, this device has the same limitations inherent in all visual observations . . . variations imposed by light intensity and background conditions. Another drawback this has is that very light smoke densities—achieved with a properly-operated engine— cannot be checked with this instrument.

A suction pump drawing smoke through filter paper also has been tried. This is available in both hand-operated and motor-driven models. The pump is designed for use with oil burners. If the manufacturer's instructions are followed, to take six strokes on the pump, the sample tabs would pick up a density far beyond the diesel oper-

ating range. Smoke from the best operating engine will appear dense since this test is sensitive enough to pick up cigarette smoke. But the number of pump strokes can be reduced and a comparison made.

Photographs have been taken of smoke. But here again the background was the big variable and completely nullified results. A change in background, advised Chairman Miller, could almost completely reverse the significance of smoke pictures.

The Committee also learned about an instrument that operates on the precipitation principle. Attached to the exhaust tail pipe, the instrument is operated from within the vehicle by squeezing a syringe. The exhaust solids trapped in the instrument are then emptied on a paper disc. They give a comparative measure of smoke density. Committee member Gil Roddewig, GMC Truck & Coach Division, felt that while this is not a laboratory instrument, it is sufficiently accurate for field work. The group will check this instrument and try to correlate its results with those using a Ringelmann Chart.

Group Probes Cures For Lead-Fouled Plugs

CHEMICAL and mechanical routes to reduction of spark-plug fouling from tetraethyl lead in aviation fuels are being studied, according to reports recently made to the SAE Ignition Research Committee.

Extent of fouling centers about the distribution of the scavenger with respect to the tetraethyl lead. Committee Chairman A. L. Beall, Wright Aeronautical Corp., says this is one reason why fuel injection engines are less susceptible to lead fouling than carbureted powerplants.

Scavengers such as ethyline dibromide and ethylene dichloride boil at temperatures several hundred degrees below tetraethyl lead. In the long trip from carburetor to engine cylinder, some of the scavenger boils off while the lead remains in the liquid state.

Injecting fuel directly into the cylinder heads, with fuel injection systems, makes for better relative distribution of lead and scavenger since the distance traveled is only from injection nozzle to combustion chamber.

Work is progressing on new scavenger compounds which have boiling points closer to that of lead.

The SAE Ignition Research Committee feels that a government agency should undertake research on lead fouling in the light of the scope and seriousness of the problem.

WILLIAM R. BECKERLE, has joined the SAE technical committee staff and will assist M. L. Stoner in the work of the SAE Aeronautics Committee, its subdivisions, and committees. Beckerle comes to SAE from the Wright Aeronautical Corp., which he joined in 1940. At Wright he worked in the experimental test department, field service, field engineering, and engineering technical department, where he was assistant project engineer. He received his mechanical engineering degree from the University of Illinois.

Pick Involute Spline Over Straight Tooth

INVOLUTE splines are favored over straight-tooth splines for hydraulic pump shafts by the Hydraulic Power Controls Subcommittee, of the SAE Construction and Industrial Machinery Technical Committee.

Discussion of pros and cons of both types at a recent meeting pointed up facts on which the group based its decision.

On the plus side for involute splines, it was said, is the fact that they are 25 to 40% stronger than straight side splines because of the wide base of their teeth. Involute splines also provide more bearing area.

Fits can be controlled more accurately because the hob can be fed deeper without affecting the involute profile. Doing this with a straight-spline hob produces undercuts at the tooth base, so that there is only about 30% bearing between mating teeth. Additionally, involute spline broaches can be shorter because teeth are shallower.

It also was argued that SAE straight-tooth splines limit the user to six, ten, or sixteen teeth. SAE Standard involute splines vary all the way from six to fifty teeth.

With involute splines, contact on at least three teeth is assured due to camming action of the teeth, regardless of spacing error. This is not true of straight-sided teeth.

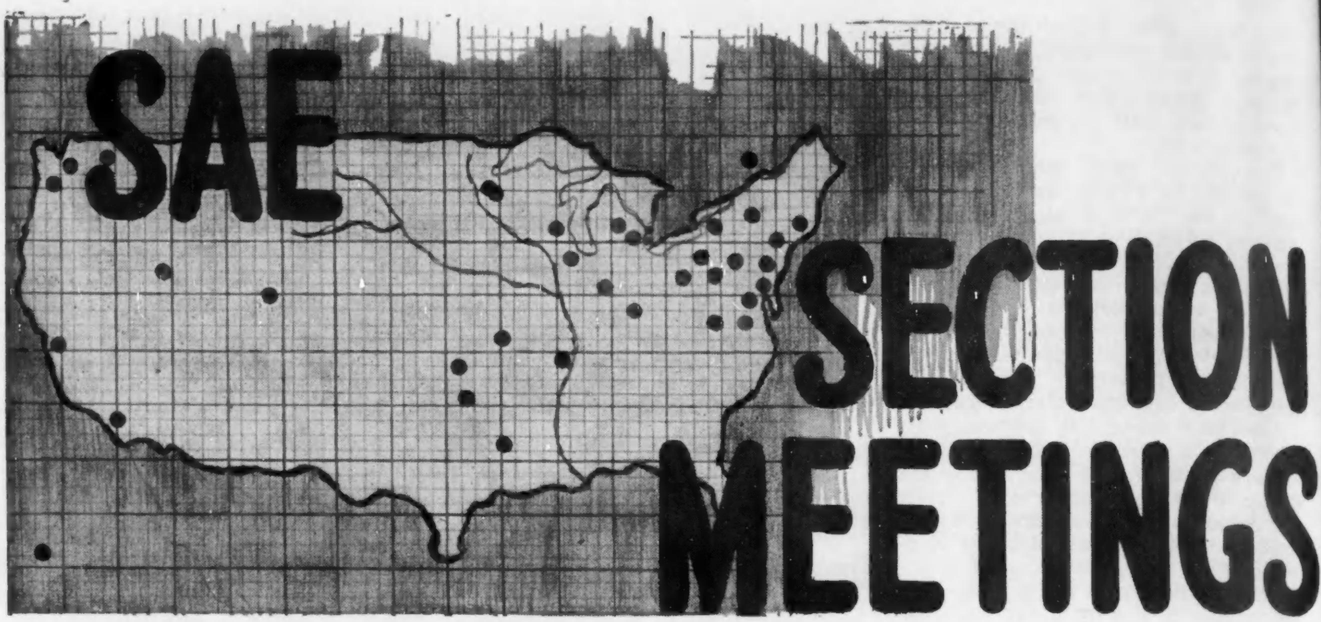
Favoring the straight splines is the fact that shop men like them because they are easy to measure. However, the straight spline hob tends to generate large fillets at the root of the male teeth, which interfere with mating teeth. Small tits are placed on the hob to eliminate these fillets. But these wear rapidly and also decrease the splined shaft's strength as they generate a slight hollow at the base of the teeth.

Straight spline broaches must be slightly tapered on teeth sides to keep from wedging and tearing the hole.

Machine tool builders are swinging over to involute splines, Subcommittee members reported, because of troubles from inaccuracies of straight splines. Several tractor manufacturers and hydraulic makers also were said to be switching to involute splines.

NATIONAL MEETINGS

MEETING	DATE	HOTEL
WEST COAST	Aug. 15-17	Multnomah, Portland, Oreg.
TRACTOR	Sept. 13-15	Schroeder, Milwaukee, Wis.
AERONAUTIC and Aircraft Engineering Display	Oct. 5-8	Biltmore, Los Angeles
DIESEL ENGINE	Nov. 1-2	Chase, St. Louis, Mo.
FUELS & LUBRICANTS	Nov. 3-4	Chase, St. Louis, Mo.
ANNUAL MEETING and Engineering Display	Jan. 9-13, 1950	Book-Cadillac, Detroit
PASSENGER CAR, BODY, and PRODUCTION	March 14-16	Book-Cadillac, Detroit
AERONAUTIC and Aircraft Engineering Display	April 17-19	Statler, New York



New England Section Holds Annual Outing

• New England Section
A. R. Okuro, Field Editor

June 24—Conspicuous interest in the annual outing was demonstrated when nearly 200 members and guests gathered at the Marlboro Country Club for a day of fun.

Golf, horseshoes, 1949 automobiles, two top-ranking local racers, and a memorable showing of veteran motor cars kept everyone occupied during the daylight hours.

The line-up of veteran cars and their owners included:

1911 Oakland, William Sylvester
1918 Locomobile, Edward Baker
1910 Stevens-Duryea, Harry Knowles
1923 Rolls Royce, Bentley Warren
1922 Stanley Steamer, Stanley Ellis
1919 Franklin, Sherman Whipple
1913 Ford Model T, Abbott Motors
1921 Mercer, Robert Townsend

An excellent steak dinner, appropriate entertainment, and the award of 120 prizes were arranged by Glenn Whitham, chairman of the outing committee and his helpers, Neal Bogren, Robert Gardner, Alfred Hunt, and Eben N. Smith.

Former Chairmen Given Paperweights

• Virginia Section
J. Y. Ray, Field Editor

June 27—Section Treasurer Percy Carr, on behalf of the Section membership, presented each of the four former chairmen with a paper weight made in the form of a pyramid. The front includes the SAE emblem, the name of the chairman, and the years during which he served.

Former chairmen honored were Jean Y. Ray, 1945-1946; Hausford B. Truslow, 1946-1947; Lucien W. Bingham, 1947-1948; Paul R. Lauritzen, 1948-1949.

Russell G. Riley discussed new design features incorporated in the engines of 1949 cars and trucks and the trends that indicated improvements to be expected in the near future. Discussion brought out the fact that operators can expect an increase in the number of manufacturers using the V-type engine and that many of these will be equipped with overhead valves. Compression ratios will also increase gradually.

Riley, who spent a month at the Indianapolis Speedway prior to the Memorial Day race working with various car crews, also reviewed some of the highlights of the race and described some of the novel ideas incorporated in the design of the cars.

At the conclusion of his talk, the speaker was presented with a Smithfield ham, a practice which the Virginia Section resumed with the end of rationing.

Membership Tours Technical School

• Kansas City Section
K. J. Holloway, Field Editor

June 14—On a tour of the Central Radio and Television Schools, Inc., SAE members viewed the school's facilities for training airline reservation personnel, flight and ground radio operators, and radio maintenance technicians.

Students performed duties of the specialized jobs for which they are training, while the methods and objectives were explained by members of the school faculty.

Hostess training was demonstrated by trainees serving dinners which were prepared, packaged, and served the same as typical airline meals, while **Howard Engstrom,** a Central navigation instructor and pilot, carried on simulated radio contact with a ground station to provide an insight to this portion of a pilot's duties while on an instrument flight.

C. L. Foster, executive vice-president and general manager of the Central Schools, emphasized the impending growth of the television industry by pointing out that as of June 7, 1949, 64 television stations were in operation, 110 additional construction permits have been granted, and 386 applications for construction permits are pending. It was estimated that approximately 2 million television receiving sets are now in operation and that this number may be increased to 3 million by the end of this year.

Foster said that WDAF, Kansas City's first television station, will probably be in operation by September 15, 1949. Transmission of a test pattern is planned for approximately August 15 to permit installation and adjustment of receivers prior to the beginning of program transmission. It is anticipated that connection of this station to a network of 28 eastern stations by means of a coaxial cable will not be completed before the end of 1950.

In discussing air navigation aids, **Fred Myers,** supervisor of aircraft radio maintenance, said that the development and installation of these aids is divided into two phases, an interim program to be completed in approximately five years and an ultimate pro-

gram to be completed in approximately ten years. In connection with the interim program, instrument landing system (ILS) installations, of which 83 are now completed, will be increased by 99, of which 61 are being constructed and 38 are in the current budget. The CAA has requested an additional 13 precision beam radar (GCA) installations to supplement the 3 now in use.

The 285 very-high-frequency omnidirectional ranges (VOR) now in use will be supplemented by 47 which are now under construction and an additional 68 which are included in the budget.

In completing the ultimate phase of this program, Myers said, it is planned to combine these and other navigational aids with television to permit presenting the pilot with additional information. This combination (teleran) is intended to aid scheduled flights with less hazard due to weather interference.

Reviews Factors in Lubricant Life

• St. Louis Section
G. C. Husbands, Field Editor

June 14—As factors affecting useful life of lubricating oils in engines, **T. P. Sands** listed temperature, crankcase ventilation, condition of engine, and type of service.

Most of these could be improved through better design and proper maintenance, said Sands, who is chief automotive engineer, Monsanto Chemical Co.

He reported on results of tests run to determine benefits of introduction of additives into lubricating oils in combating corrosion, increasing life and reducing consumption.

Announce Winners of Paper Contest

• Washington Section
Hoy Stevens

June 29—First prize of $50 in the section's first student paper contest will go to Frank Martin, Jr. of the University of Maryland for his paper "The Hot-Air Engine as an Automotive Prime Mover," the governing board, which judged the contest, announced.

Second prize of $25 goes to Jay W. Miller, also of the University of Maryland, for "Certain Design Problems of the Rear-Engine Automobile." And third prize of $15 was won by Herbert Marsteller, George Washington University, for "Need for Better Street Lighting." Harry R. Smith, Jr., University of Maryland, received honorable mention.

The contest was open to undergraduates of the colleges in and around Washington. Most of the 14 papers submitted were term papers; theses written for degree requirements were ineligible. Subjects were limited to "automotive" topics, but judges gave broad interpretation to "automotive."

Besides the cash prizes, the section will pay for student enrollment in SAE for the winners. Plans call for presentation of the awards at the section's October meeting.

Hawaii Leads in Air Travel

• Hawaii Section
Rene Guillou, Field Editor

June 20—That 83% of the Hawaiian population travels by air each year was stated by **William Holloway**, safety engineer at the Hawaii Aeronautics Commission.

This is believed to be a world record. Travel between the islands, which is chiefly by air, accounted for 376,000 air passengers in 1948, while 93,000 flew between Hawaii and the mainland. Over 22,000,000 lb of air cargo were also carried within Hawaii in 1948. Of this total, 35% originated at Honolulu, while almost 60% of the cargo terminated there, including large shipments of agricultural products from the outer islands, Holloway said.

The 15 airports under jurisdiction of the Hawaii Aeronautics Commission actually include more paved area than do all the highways of the Territory. Holloway described the various maintenance and service activities of the Commission, and particularly their safety measures. The life of a passenger has yet to be lost on scheduled flight between the islands.

The meeting was closed with a showing of the Civil Aeronautics Authority film "Our Town Builds an Airport."

Foresees Growth of Stratocruiser

• Metropolitan Section
John D. Waugh, Field Editor

June 16—The Stratocruiser is a big transport now, and it is likely to grow

in gross weight, said **D. B. Martin**, Boeing's New York representative.

His statistics, plus a short motion picture showing the Stratocruiser's features, convinced the audience. According to Martin, the Stratocruiser is now flying at a certificated gross weight of 142,500 lb, powered with four 3500-hp engines. Its fuel capacity is 7790 gal and maximum cruise speed 340 mhp.

With normal rate of development of powerplants and propellers, the Stratocruiser may some day operate at its structurally limited gross weight of 153,000 lb. Should the necessity for greater weight arise, gross weight can be raised through modification of wings and landing gear, Martin said.

Noting that bigness seems to inspire passengers' confidence, one airlines engineer predicted that the Stratocruiser's size will win it public acclaim.

Other discussers pointed out contributions made by suppliers in this area to the Stratocruiser's success, citing the Dehmel flight simulator used in training crews and a new fuel nozzle developed to eliminate splashing as fuel is fed into the airplane's tanks at 200 gal per min.

Urges Conservation of Fluid Fuels

• Pittsburgh Section
Murray Fahnestock, Field Editor

May 28, 1949—Speaking on the "Possibilities of Future Fuels," **R. J. S. Pigott**, past-president of SAE, said that proved reserves of petroleum have remained fairly constant at somewhat over 10 years for some time, due to discoveries and improved recoveries in old fields; but that this cannot continue for long as now most of the shallow pools have been found and an increasing part of the discoveries are at 10,000 ft or deeper.

For the past generation, we have been improving methods of total recovery by such means as flooding with water to push out the oil, and repressuring with gas produced with the oil and recovered from the separators, Pigott said.

Any improvement in engine efficiency, such as raising the compression ratio, tends to conserve fuel. While a 12.5-compression ratio would give an operating gain of 35 to 40% compression over ratios of 6.5; the experimental engines so far built need 100 octane research fuel, not yet available, and not likely to be, in large volume, Pigott explained.

As petroleum becomes scarcer and more expensive, we may reduce the 3.8% burned in central stations for production of electric power and the 10.3% of natural gas, and could thus

add about 10% to fuel available for automotive uses.

Consequently, designers should consider the transfer of all energy use possible from liquid to solid fuel reserving the liquid and gaseous fuels for those operations that cannot do without them.

Rescue Advances

Cont. from p. 72

frequencies still are a serious requirement.

As for the rescue itself, the pick-up of survivors in difficult terrain will depend upon future capabilities of the helicopter. Even the pick-up of a survivor by a surface craft at sea has its dangers, especially if he is injured, since an effective and safe litter and hoist is not yet available.

Aside from these shortcomings, this fact is becoming increasingly clear: Search and rescue is the king of unmentionables to the transportation industry's public relations man—to be praised in private, and ignored in public. There seems to be an unexpressed fear that public education in survival is an admission that an accident might happen.

Let us no longer speak in whispers of search and rescue. Modern common carriers are safe; but there are calculated risks in travel. The average traveler acknowledges this risk when he buys trip insurance. He should expect, if not demand, that he be given some briefing on evacuation and survival methods in case of a disaster. (Paper "Developments in Search and Rescue," was presented at SAE National Aeronautic and Air Transport Meeting, New York, April 11, 1949. This paper is available in full in multilithographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

Experts Pool Know-How On New Air Transports

HOW to get from modern airliners the life and performance built into them is told by 13 engineers in the symposium "Getting the Best Out of Our New Transports."

Eleven design engineers give tips about their equipment, distinguishing between abusive and proper usage. They also show how inter-airline standardization would permit them to produce less costly equipment. Two airline operators reveal what they consider sound operating and maintenance

practices. And they indicate which components need improvement and where standardization would help the operator.

The design engineers who contributed to this symposium and the subjects they cover are: P. C. Scofield, AiResearch Mfg. Co. . . . heating, ventilating, air conditioning, and pressurization; E. S. Gallagher, General Electric Co. . . . electrical system; R. A. Brown, Minneapolis-Honeywell Regulator Co. . . . automatic equipment; James Robinson, Vickers, Inc. . . . hydraulic equipment; P. D. Doran, Pratt & Whitney Aircraft Engines . . . engines; C. F. Baker, Hamilton standard Division, United Aircraft Corp. . . . propellers; H. G. Tarter, Bendix Products Division, Bendix Aviation Corp. . . . carburetors; W. V. Hanley, Standard Oil Co. of Calif. . . . fuels and lubricants; A. E. Raymond, Douglas Aircraft Co., Inc. . . . airplanes; R. W. Young, Wright Aeronautical Corp. . . . powerplants; D. E. Fritz, Jack & Heintz Precision Industries, Inc. . . . equipment.

The two airline operators who authored sections of the symposium are Charles Froesch, Eastern Airlines, Inc., and William Littlewood, American Airlines, Inc. (The symposium "Getting the Best Out of Our New Transports," was presented at SAE National Aeronautic and Air Transport Meeting, New York, April 12, 1949. This symposium is available in full in multilithographed form from SAE Special Publications Department. Price: $2.00 to members, $4.00 to nonmembers.)

Search Never Ends For Better Tires

Based on paper by

DR. ARTHUR W. BULL

U. S. Rubber Co.

EXTENSIVE research proceeds continuously on the seemingly simple, yet rugged motor vehicle tire.

Toughness of a tire is evidenced by the rigid specifications its components must satisfy:

1. Bead wires must withstand all loads transmitted to them by the thousands of cords anchored at the beads.

2. Tire cords must have great strength, flexibility, some stretch, and high fatigue resistance since they are subjected to millions of stress reversals and elongation changes. They must function under temperatures ranging from −40 to 300 F.

3. The tire breaker, which incorporates additional cord or fabric layers directly under the tread, must resist rupture or bruise when the tire runs

over large stones or other obstacles.

4. The thread weather-proofs the carcass to protect the cords; provides frictional contact with the road through which all steering, braking, and driving forces are exchanged between car and road; and must have superlative resistance to abrasion.

Although tires seem quite simple, operation of such a flexible structure is highly complicated. Every detail of material and construction has been subjected to increasingly critical scrutiny for many years.

For example, to build more miles per dollar into tires, they go through grueling lab and field tests. Hundreds of tires are pounded to destruction in a testing laboratory by continuous running before bringing out a new tire type. On vehicles they are tested in a variety of climates and over different roads—ranging from Duluth to Death Valley.

In 1948 we tested more than 14,000 tires and tubes over a total of 220 million miles. This equals 7000 to 8000 trips across the country by one vehicle.

Tire advances over the last 25 years are said to be worth at least $50 billion to American motor vehicle users. (Paper "Tire Problems," was presented at SAE Western Michigan Section, Muskegon, Feb. 17, 1949. This paper is available in full in mimeographed form from SAE Special Publications Department. Price: 25¢ to members, 50¢ to nonmembers.)

New Members Qualified

These applicants qualified for admission to the Society between June 10, 1949 and July 10, 1949. Grades of membership are: (M) Member; (A) Associate; (J) Junior; (Aff.) Affiliate; (SM) Service Member; (FM) Foreign Member.

Baltimore Section

R. D. Austin, Jr. (J), Clarence M. Rusk (M), R. H. Sheppard (M).

British Columbia Section

Albert Henry Bennett (A), Ronald H. Carter (A), Keith Forbes (A), George Arthur Lloyd (J), Thomas Walsh McKee (A), Capt. Roy S. Minter (A), Perry Moore (A).

Buffalo Section

Charles E. Blakeslee (A), Howard J. Jansen (J), Ed. C. Merry (M).

Canadian Section

Lee Tolmie Craig (A), Frank J. Crothers (A), A. Reg. Dale (A), Z. M. Ferley (J), Donald Lachlan McGillivary (M), Douglas M. Park (A), Harry T. Smith (A), W. C. Smith (A).

Central Illinois Section

Harlan Banister (J), Edgar S. Cheaney (J).

Chicago Section

Alfred G. Bartlett (A), Robert E. Bourke (M), Joseph J. Brett (J), Addison C. Hoof (J), Evan E. Olmstead (M), Charles W. Ricker, Jr. (M), Howard B. Scoville (M), John Seifner, Jr. (M), Norman N. Snyder (M), Roy H. Stevens (A), Donn Sutton (A), Boyd A. Whitehead (M), Arnold F. Zimmerman (A).

Cincinnati Section

C. Marvin Dorsey (A), Joseph Graber (A), Wayne D. Van Atta, Jr. (J).

Cleveland Section

Malcolm Graham Anderson (M), Richard S. Brosius (M), F. L. Hall (M), Samuel G. Laughlin (A), Paul E. Lewis (M), Paul A. Miller (M), W. R. Myers (A), Edward Charles Powell (J), Rudolph J. Schonitzer (J).

Dayton Section

Kenneth C. Baker (A), Clarence H. Lorig (M), Robert H. Lundberg (M), Glen E. Mealy, Jr. (A), Rudolph E. Olila (M), S. R. Prance (M), Charles Edwin Robertson, Jr. (J), Joseph E. Scandling (M), Richard Kelsey Sullivan (A).

Detroit Section

Hugh H. Benninger (M), Russell A. Blanchard (A), Harold Raymond Boyer (M), H. Clifford Brosey (J), J. S. Decker (M), Philip R. Denham, Jr. (J), Elmer J. Engelman (M), Earle H. Gould (M), Francis William Holland (A), Alexander Hossack (J), Fred W. Hottenroth (M), W. G. Kinmont (A), Eugene W. Konrad (M), Chris T. Kope (M), Robert S. MacDonald (J), Marvin Richard Marsh (J), Philip Mazziotti (J), Charles Donal Moore (J), Carl A. Olson, Jr. (M), Willard R. Phillips (M), Michael Pinto (A), Clement W. Prestler (A), E. R. Reeves (A), Helene A. Rother (A), Charles M. Rubley (M), Ronald W. Scheck (J), Ralph N. S. Scheidt (J), George J. Scranton (M), Frederic R. Smith (A), Guy L. Stevens (M), Kenneth L. Vogt (M), Clifford R. Warg (J), Robert W. Wolfe (M), H. O. Woods (A), Frank Winston Wylie (A).

Hawaii Section

David J. Miyashiro (A).

Indiana Section

Harrold J. Bates (M), W. E. Brown (M), Richard W. Dickey (J), John Henry Glass (M), Robert L. Kessler (M), Milton L. Munson (J).

Kansas City Section

Francis Lee Spruill (M).

Metropolitan Section

Thomas J. Greene (A), Robert S. Grier (A), L. E. Grubb (M), J. Robert Hinig (A), Hugh S. Kelly (J), Peter J. Schuessler (A), Edwin Henry Snyder, Jr. (J).

Milwaukee Section

George Colburn, Jr. (J), David Frank Cosgrove (J), Francis W. Doyle (M), James W. Martin (J).

Mohawk-Hudson Group

Dwight C. Hannay (A), Harold R. Sennstrom (M).

New England Section

George Thomas Brown (M), James A. Fleming (A), Robert M. Wallace (A), John Elijah Whipple (M).

Northern California Section

Carl R. Bergman (A), John L. Cooley (M), Paul C. Jaeger (A), E. Lloyd King (A), John Joseph Riordan (M), Theodore C. Scow (A).

Northwest Section

M. E. Earnheart (A), Phil Howe (A), Richard P. Look (J), Charles J. Wilhite (A).

Oregon Section

Newell T. Palmer (A).

Philadelphia Section

Frits Jonker (M), Daniel L. Pastell (J).

Pittsburgh Section

William W. Beck (J), J. Preston Flaherty (M), Francis C. Mueller (A), James Spence (A).

St. Louis Section

J. Kenneth Craver (A), Eugene Edward Wallace (J).

Salt Lake Group

Pete J. Visser (A).

San Diego Section

Oliver B. Lyons (A).

Southern California Section

Robert E. Douglas (J), Norman A. Faber (M), Edward H. Farmer (M), Robert Bruce Kelly (J), LaVerne Morgan (J), H. J. Schlarb (A), Manton Leo Sheehan (A), Billy L. Stocking (A), Charles Robert Thomson (A), John W. Windle (A), Walter A. Woron (J).

Southern New England Section

John Jackson Blessley (J), George Gilbert Hammarlund (M), Robert B. Powell (M).

Spokane Intermountain Section

Charles Howard Franklin (A), George W. Shields, Jr. (A), Lt. Comdr. Herbert James Ward (SM).

Syracuse Section

Jack Frederick Fralick (M).

Texas Section

Wayne E. Patterson (J).

Twin City Section

H. L. Harms (M).

Virginia Section

Archie L. Grinels (A), John A. Mahrley (A).

Washington Section

Robert Henry Bright (M), William Frederick Spurr (M).

Western Michigan Section

Jacob O. Funkhouser (A).

Wichita Section

John D. McClurkin (M).

Outside of Section Territory

B. E. Derry (A), James Vernon Doig (A), Yale W. Ehret (M), John Edward Gilmore, Jr. (A), Cedric E. Hulme (M), Richard Howard LeTourneau (J), Charles M. Marlatt (J), Ernest Sewell, Jr. (A).

Foreign

Paul Wilhelm Aberg (FM), Sweden; Andre Cattin (FM), France; Reginald Bruce Godfree (FM), England; Maurice Carne Hall (FM), India; William Thomas Vernon (FM), England; John Walter Wells (A), So. Africa; Edward Brian Wilson (FM), England.

Applications Received

The applications for membership received between June 10, 1949, and July 10, 1949 are listed below.

Baltimore Section

Carl Kenneth Kummerlowe.

British Columbia Section

Frank M. Francis, Francis Hugh Hymers.

Canadian Section

James E. Gordon, John Joseph Payne, Lt. Walker James Power, Willard James Russell.

Chicago Section

Arthur E. Brown, Victor R. Clark, Milo F. Denick, Franz Richard Hildebrecht, C. L. Sadler, Frits Henry Sandberg, Floyd J. Teed, Wymer G. Vogel.

Cleveland Section

Howard H. Arnold, Raymond C. Firestone, Mahlen Francis Kahler,

William E. Kilgore, Alvah Orlando Osborn, Dan F. Ostrowski.

Dayton Section

M. E. Rasper, James Kenneth Thompson.

Detroit Section

William M. Barr, Ross William Christian, J. D. Collins, Forrest Worth Cook, Joseph F. Csernai, Frederick F. Denny, Herbert R. Fortgang, Frederick J. Graham, Charles F. Gray, Saifi T. Haidery, Richard K. Harwood, Thomas O. MacDonald, Eugene Charles Molnar, Henry Monik, Robert M. Morkin, John W. Pyuro, Charles Richter, Jr., Arthur John Slemmons, David Karl Wirth.

Hawaii Section

William Lucas Ford, Jr.

Indiana Section

Ben H. Crews, Thomas G. Driscoll, Eugene Phillip Welcher.

Kansas City Section

Ernest L. Bailey.

Metropolitan Section

Bruno Butti, Frank D. Idoni, Sidney Lanzet, Louis Daniel Rand, Albert Saliano, Arthur Stutzer, Jr., George Weinstein.

Mid-Continent Section

Albert N. Janco, Roy S. Neff.

Milwaukee Section

Leonard R. Risse, Carleton Harry Swanson.

Mohawk-Hudson Group

W. Clement Palin, Jr.

New England Section

Herbert England Jacques, Russell L. Keene.

Northern California Section

Robert E. Busby, Charles Thomas Coleman, William J. Drummy, III, Henry F. Kleemyer, Wilfred J. Lamorey, Jr., W. W. Schuldt, Sidney D. Selan, Samuel Weiss, Tilak RajLall.

Northwest Section

R. G. Fender, Michael Guidon, III, Robert William Jeffs.

Oregon Section

Roscoe Wilson Clarke, William A. Cramer.

Philadelphia Section

Robert Burke Benham, Ralph W. Benson, Wesley R. Frysztacki, Theodore R. Gensel.

Pittsburgh Section

Thomas N. McVetty.

St. Louis Section

John Richard Mulin.

Salt Lake Group

Harold E. LaBelle, Jr.

Southern California Section

Harold T. Fuller, Roland E. Gagon, Sam M. Kelly, Orin "C" Lamb, Robert Arnold Norvell, Herbert L. Podell, James Edward Stadler, Robert L. Vandeveer.

Southern New England Section

William Crampton Becker, John L. Morrissette, Russell E. Phelon, Joseph F. Regan.

Spokane-Intermountain Section

Claude J. Douty.

Texas Section

Vincent Paul Citrullo.

Twin City Section

William N. Blatt.

Virginia Section

E. H. Marsteller, Jr.

Washington Section

Arthur Stone.

It's HYATT

THE Australian-built "Holden" is an innovation in automotive transportation for the land "down under". The "Holden" car is built by and for Australians. Ninety-five percent of its parts are made at home, with roller bearings one of the few units produced outside the Commonwealth—and they are Hyatts.

Hyatt Roller Bearings were chosen by Holden because of Hyatt's proven performance in automotive equipment—notably their high load carrying capacity and longer life.

Through the use of the finest steels, special Hyatt heat treating methods, precision grinding, accurate inspection and modern assembly procedures, Hyatt Roller Bearings have acquired worldwide recognition for their always dependable quality. Hyatt Bearings Division, General Motors Corporation, Harrison, New Jersey; Detroit, Michigan.

HYATT ROLLER BEARINGS

Williamsport Group

Samuel Dominick Greco.

Outside of Section Territory

Glenn W. DeVault, Kenneth Glade, L. N. Gushard, Robert F. Knox, Roland Earl Luecht, Harold Edward Meunier, Monroe L. Patzig, Karl Siegle, Captain John Smith, Jr., Raymond A. Weigel.

Foreign

Alan Reeve Croft, England; William Milne, S. Rhodesia.

Change of Address

So that your SAE mail will reach you with the least possible delay, please keep SAE Headquarters and the Secretary of your local Section or Group advised of any changes in your address. Such notices should be sent to:

1. Society of Automotive Engineers, Inc., 29 West 39th St., New York 18, N. Y.
2. The Secretary or Assistant Secretary of your Section or Group at the addresses listed below:

Baltimore

Ward L. Bennett, Baltimore Transfer Co., Monument & Forrest Sts., Baltimore 2, Md.

British Columbia

Burdette Trout, Suite 6, 1436 Pendrill St., Vancouver, B. C., Can.

Buffalo

Robert W. Morgan, 32 Windover, Hamburg, N. Y.

Canadian

Frank G. King, Canadian Automotive Trade, MacLean-Hunter Publishing Co., Ltd., 481 University Ave., Toronto 2, Ont., Can.

Central Illinois

James C. Porter, 120 Isabel Ave., Peoria, Ill.

Chicago

Floyd E. Ertsman, Chicago Section, SAE, 1420 Fisher Bldg., 343 S. Dearborn St., Chicago 4.

Cincinnati

B. E. Bennett, 55 Park Ave., Elsmere, P. O. Erlanger, Ky.

Cleveland

(Miss) C. M. Hill, 7016 Euclid Ave., Cleveland 3, Ohio

Dayton

Fred Schwenk, Fred's Auto Hospital, 1522 Xenia Ave., Dayton 10, Ohio

Detroit

(Mrs.) S. J. Duvall, Detroit Office, SAE, 100 Farnsworth Ave., Detroit 2, Mich.

Hawaii

Fred Hedemann, 2110 A Kakela Pl., Honolulu, T.H.

Indiana

Charles K. Taylor, L.G.S. Spring Clutch Corp., Indianapolis 6, Ind.

Kansas City

John P. Dranek, Box 304, Grandview, Mo.

Metropolitan

(Miss) J. A. McCormick, Society of Automotive Engineers, 29 West 39th St., New York 18, N. Y.

Mid-Continent

Delton R. Frey, 1427 S. 20th St., Chickasha, Okla.

Milwaukee

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Salt Lake

Stanley W. Stephens, 36 E. Crystal Ave., Salt Lake City 6, Utah.

Williamsport

Carroll S. Townsend, General Armature & Mfg. Co., Lock Haven, Pa.